KALINGA INSTITUTE OF INDUSTRIAL TECHNOLOGY

BHUBANESWAR, ODISHA, INDIA

PROJECT REPORT

ON

TORQUE REVERSAL GEARBOX

UNDER THE ESTEEMED GUIDANCE OF

PROF. ISHAM PANIGRAHI

| SCHOOL OF MECHANICAL ENGINEERING |

ACKNOWLEDGEMENT

It is our privilege to express our sincerest regards to our project co-ordinator **PROF. ISHAM PANIGRAHI**, for his valuable inputs, able guidance, encouragement, whole-hearted cooperation and constructive criticism throughout the duration of our project.

We deeply express our sincere thanks to the Dean, School Of Mechanical Engineering, **PROF. (DR.) K.C. SINGH** for encouraging and allowing us to present the project titled **“TORQUE REVERSAL GEARBOX”** at our department for the partial fulfilment of the requirements leading to the award of B.Tech degree.

TEAM MEMBERS

Serial No.	Name	Roll No.
1	A.NAVEEN	902001
2	ABHIJEET KUMAR SAHU	902003
3	ABHIJIT MANDAL	902004
4	ABHINAV KUMAR	902007
5	ABHINAV KUMAR	902008
6	ABHINAV PADHY	902009
7	ANSHUMAN PARIDA	902039
8	ARUN SINGH YADAV	902046
9	ARUNIM BHATTACHARYA	902047
10	ASHISH KUMAR	902051
11	UJJWAL SENGUPTA	902153

CERTIFICATE

This is to certify that the following students of Final Year B. Tech (Mechanical Engineering) of School of Mechanical Engineering at KALINGA INSTITUTE OF INDUSTRIAL TECHNOLOGY, BHUBANESWAR have undergone project work on "Torque Reversal Gear Box" in 8^{th} Semester. They have successfully completed the project assigned to them.

The project report entitled "Torque Reversal Gear Box" embodies the original work done by them during the above period under guidance of the undersigned.

I wish them all success in their future endeavours.

Prof.(Dr.) K.C.Singh
Dean
School of Mechanical Engineering
KIIT University, Bhubaneswar

Prof. Isham Panigrahi
Assistant Professor
School of Mechanical Engineering
KIIT University, Bhubaneswar

CONTENTS

ABSTRACT

Since the World War 2 there has been a tremendous increase in privately owned vehicles, which has triggered a competition for space .A lot of time is spent in changing the direction of movement of these vehicles. A lot of time drivers get stuck on the road unable to reverse their direction of motion which is mainly due to lack of space for turning around like in narrow lanes. Solution to this problem can be found on the age old military tank-technology i.e. the ability to rotate about its own center. So we have thought of porting this technique to civilian vehicles. The solution is essentially a gearbox which has the capability to transmit motion both in forward and reverse direction. This is intended to be just an add-on for the existing drive train. The main focus here is to turn a set of wheels (either the right set or the left set) in the same or opposite direction. This kind of motion imparts an overall rotation to the vehicle which effects in turning it through any angle with zero turn radius. The gear box is essentially a set of gears and clutch arrangement which works in complete unison to bring about the required effect.

NOMENCLATURE

Gears	
σ_{ut}	Ultimate tensile strength
M	module
P_t	Maximum tangential force
$P`_t$	Rated tangential force
Z	Number of teeth
z`	Number of teeth on virtual spur gear
d,d`	Pitch dia on helical and virtual spur gears
d_a, $d_a`$	Addendum dia of helical and virtual spur gears
α_n, Ψ_n	Pressure angle and helix angle
Shaft	
σ_{ut}	Ultimate tensile strength
σ_{yt}	Yield strength
D	Shaft diameter
P_t , P_r ,P_a	Tangential ,radial ,axial load
k_b , k_t	Shock and fatigue factors
Conical Clutch	
P	Axial Applied force kW
M_t	Clutch Torque (Nm)
M	Coefficient of Friction
D, d	Rear, Frontal diameter of clutch
A	Semi-cone angle
Spring	
P	Axial force
D	Outer diameter
D	Wire diameter
K	Spring stiffness
C	Spring Index
G	Modulus of rigidity
Keys	
L	Length of key
B	Breadth of key
H	Height of key
FOS	Factor of safety

3D REALISTIC RENDERING OF THE MODEL DESIGNED FOR THE PROJECT

CHAPTER-1

INTRODUCTION

1.1 History and development of torque reversal gearbox

French inventors Louis-Rene Panhard and Emile Levassor are credited with the development of the first modern transmission. They demonstrated their three-speed transmission in 1894 and the basic design is still the starting point for most contemporary manual transmissions.

Improved design

Panhard and Levassor used a chain drive on their original transmission. In 1898 auto maker Louis Renault used their basic design, but substituted a drive shaft for the drive chain and added a differential axle for the rear wheels to improve performance of the transmission.

Time Frame

By the beginning of the 20th century most cars manufactured in the United States featured a non-synchronized manual transmission based on the Panhard/Levassor/Renault design. The next major innovation occurred in 1928 when Cadillac introduced the synchronized manual transmission, which significantly reduced gear grinding and made shifting smoother and easier.

Early transmissions included the right-angle drives and other gearing in windmills, horse-powered devices, and steam engines, in support of pumping, milling, and hoisting.

Most modern gearboxes are used to increase torque while reducing the speed of a prime mover output shaft (e.g. a motor crankshaft). This means that the output shaft of a gearbox rotates at a slower rate than the input shaft, and this reduction in speed produces a mechanical advantage, increasing torque. A gearbox can be set up to do the opposite and provide an increase in shaft speed with a reduction of torque. Some of the simplest gearboxes merely change the physical direction of power transmission.

Many typical automobile transmissions include the ability to select one of several different gear ratios. In this case, most of the gear ratios (often simply called "gears") are used to slow down the output speed of the engine and increase torque. However, the highest gears may be "overdrive" types that increase the output speed.

The modern tank is the 20th century was the realization of the innovations in internal combustion engine, armour plate, continuous track, and the torque reversal gearbox. In case of tanks and some robots/rovers, the vehicle is turned by rotating one set of wheels (either left or right) keeping the other set in still position. This technology has been used to design the proposed torque reversal gearbox where both the set of wheels (left and right) rotate in opposite direction resulting in torque multiplication.

Gearbox is an integral part of transmission line. The gearbox is a power transfer device whatever it is a MT or AT. It allows the take-off of the vehicle, the transfer of the engine power to the driving wheels, in forward and reverse direction, in accordance to the driver demand and it achieves an adequate distribution of the torques and rotation speeds between the right and left driving wheels.

To execute those functions, the device is composed of:

- A coupling/decoupling device (e.g. clutch or converter)
- A range of reduction rations (e.g. gears or pulleys)
- A device allowing to change ratios (e.g. synchronizers, hydraulic actuators, …)
- A power distribution device (the differential)
- A connection with the wheels respecting their degrees of freedom (joints and drive shafts)
- A connecting device with the driver (knob, pads, …)

1.2 BASIC OUTLINE OF THE PROJECT

This project report consists of 12 chapters in which theory regarding design of Helical Gearbox and its components is dealt with. In the next chapter the history and background of this project is detailed referring various journals. In the consequent chapter the design and calculations of the components has been mentioned. The model and 2D and 3D diagrams have been shown for better visualisation of the project. In the further chapters the conclusion and future aspects of this proposed idea is portrayed. Finally the Bibliography and references are enumerated.

1.3 OBJECTIVE OF THE PROJECT

The objectives of the project are:-

- To create a torque reversing gearbox for civilian vehicles
- To increase drive control ability by reducing the turning radius
- To enable the vehicle to park using minimum road space

CHAPTER-2

GEARBOX

2.1 What is the gear box how does it transmit the power?

Gears are used for increasing the torque of the source of rotary motion having high angular momentum and low torque. This high torque is necessary for performance of work. This phenomenon of increase in torque is called gear reduction and is brought about by coupling of a smaller gear called the pinion with a larger gear. This results in reduction of torque at the expense of angular momentum. Such a gearbox is called a reducer. One more application of gears is to change the axis or plane of rotary motion with or without gear reduction.

When you open a gearbox you will see that the inner construction is very simple. Inside you will find two gears coupled with one another. The gears may be of spur, helical, cycloid, worm or bevel type. In case of gear reduction, the diameter of the output is larger than that of the input gear. If only a change in direction is required, the size of the gears is the same. Spur gears are used for heavy load but are noisy. Helical gears are preferred as they are silent in operation due to gradual engagement. If change of plane of rotation is required, hypoid gears are used.

The gear may be either of metal or plastic. This entire arrangement is enclosed in metallic or plastic housing. The point of contact of the gear teeth is well lubricated with gear oil. The gear oil must be very clean and free of abrasive materials to avoid wearing of the gears.

A gearbox is used in turbines, windmills, grinders, etc. to change the direction of the rotary motion. In automobiles a gear box is used for transfer of to power of the engine to the wheels through a differential.

In tandem with the conventional gear we are using an adjunct gearbox to add to the functionalities of the drive-train . The main add-on of this secondary gearbox is to provide turnings of zero radius. This has been inspired by the movement of military tanks . The idea from the military tanks has been utilised to design this non reducing gearbox.

Our gearbox contains a set of gears and clutches. Clutches are instrumental in determining the direction of output torque. At a given instant only one the two clutches will be in action and the other will remain idle . This arrangement helps us to attain a transmission which gives us the capability to control the individual torque transmission at individual wheels.

2.2 GEARBOX LAYOUT

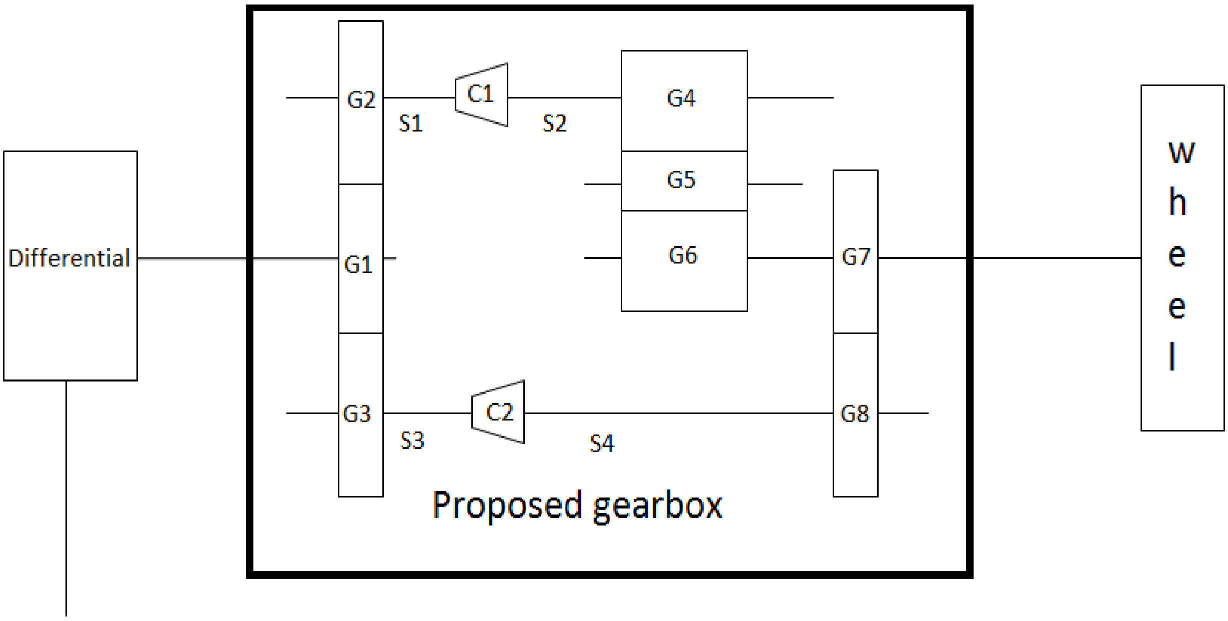


Fig.1 General layout of the gearbox

This is the general layout of the proposed gearbox . It consists of a set of helical gears and two conical clutches . The different gears and clutches has been numbered for easy reference . The driveline is same as that of a conventional vehicle , the only addition is a secondary non reducing gearbox , situated in between the differntial and the wheels . Rear axle is connected to wheels via this gearbox . The axle from the differntial is connected to the gear G1 which is mounted on the main shaft . Gears G2 and G3 mounted on the auxiliary driving shafts (S1 and S2) are in mesh with gear G1. There are two clutches C1 and C2 which are mounted on the auxiliary driving shafts (S1 and S3) . The torque transmission to the driven auxiliary shafts depends on the clutch engagement or dis-engagement . Gears G4 and G8 are then connected to these auxiliary driven shafts (S2 and S4) . The overall gear ratio is maintained to be 1:1 such that there is no speed reduction through the gearbox . Gears G4 and G6 are in mesh with the help of reversing gear G5 . The shaft containing gears G6 and G7 is the final output shaft connected to the wheels. The arrow heads shows the direcrion of transmission of torque . Gears G1 G3 G8 G7 are connected with the help of clutch C2 and this constitutes the forward transmission . Similarly gears G1 G2 G4 G5 G6 G7 are connected with the help of clutch C1 and this constitutes the reverse transmission . The configuration of the gearbox when it is transmitting torque in forward and reverse direction is shown below

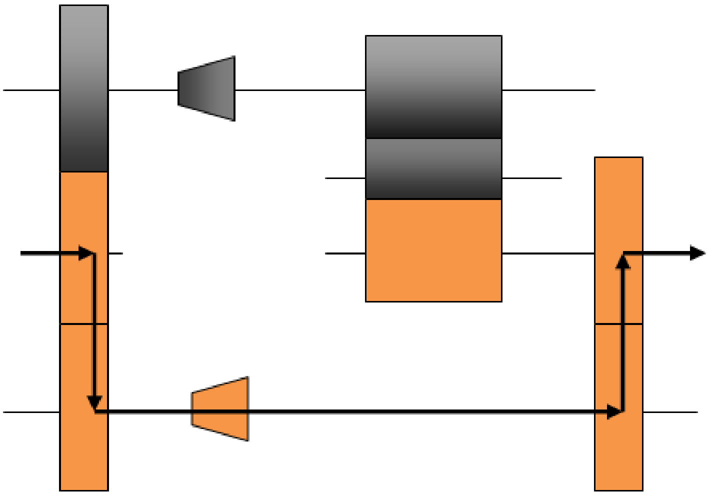


Fig.2 Gearbox engaged in forward transmission

When clutch C2 is engaged , the gears G1 G3 G8 G7 gets meshed together . From the diagram it is clear that the gear G1 and G7 will rotate in same direction when engaged . So this configuration leads to the forward transmission of torque without any speed reduction since the gear ratio maintained is 1:1 .

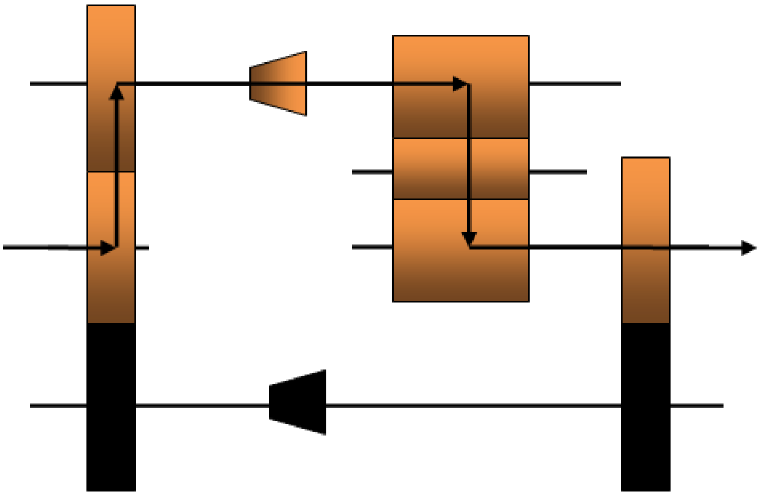


Fig.3 Gearbox engaged in reverse transmission

When clutch C1 is engaged , the gears G1 G2 G4 G5 G6 G7 gets meshed together . From the diagram it is clear that the gear G1 and G7 will rotate in opposite direction when engaged due to the presence of the reversing gear G5. So this configuration leads to the reverse transmission of torque without any speed reduction since the gear ratio maintained is 1:1 .

The gearbox is attached on the either set (left or right) of wheels . These configurations gives us individual control of wheels which helps us to obtain the required direction of transmission of torque . This helps us to achieve the required objective of turning the vehicle with zero radius .

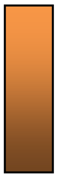
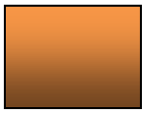
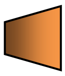
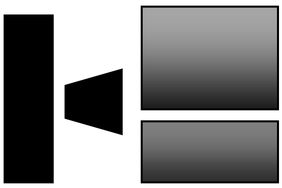
ENTITY	NAME	FUNCTION
	Helical gear 60mm dia	This gear is transmitting torque
	Helical gear 60mm dia	This gear is transmitting torque
	Helical gear 60mm dia	This gear is transmitting torque
	Clutch arrangement	Clutch engaged
	-	Inactive components i.e. not transmitting torque
	Shaft	-
	Direction of engaged components	-

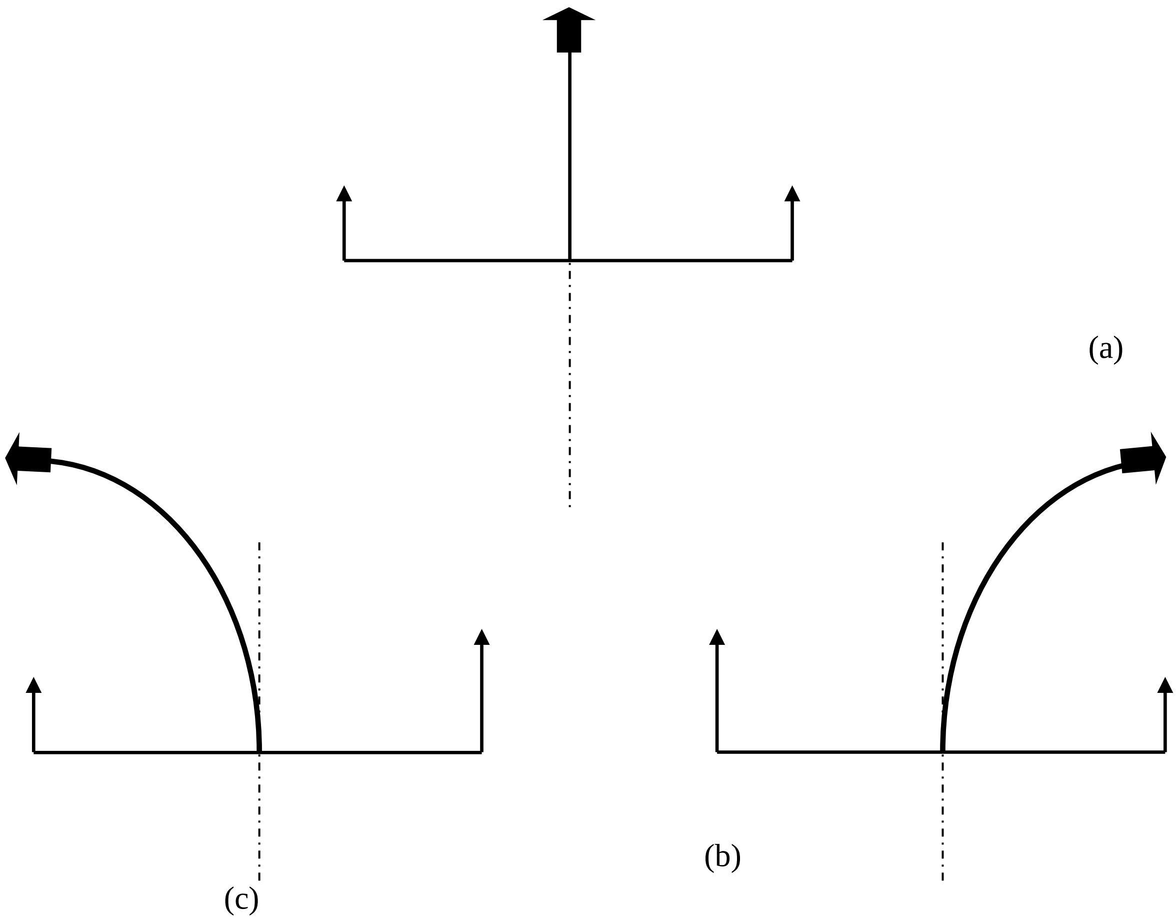
Table 1

CHAPTER-3

PROPOSED GEARBOX

3.1 WORKING PRINCIPLE

During turning in conventional vehicles the direction of torque transmission in both the set (left and right) of wheels remain same but there magnitude differs . This difference in the magnitude of torque give rise to a resultant torque and the vehicle turns in the direction of the resultant torque .



Resultant torque due to the variable speed at both set of wheels

(For conventional vehicle)

⟶ Represents the direction and magnitude of torque

Fig.4

If direction of torque transmission in both the set (left and right) of wheels are made opposite in direction keeping their magnitude same then the resultant torque produced will be extremely large compared to the torque produced in the case of conventional vehicles . This helps the vehicle to turn instantly keeping it's turning radius minimal . The direction of the torque transmission is controlled by the secondary gearbox . This setup will aid the driver comfortability while turning at the sharp turns and also in the narrow lanes where the turning radius available is very less . This setup will also aid in the parking of the vehicle .

3.2 Difference between Conventional and Proposed Gearbox

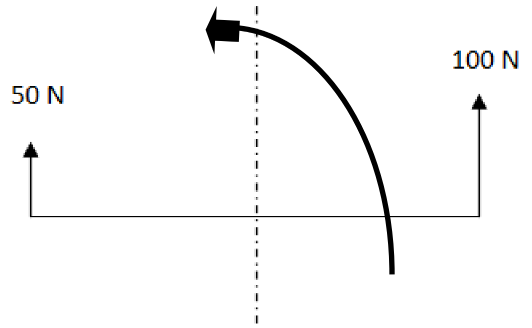
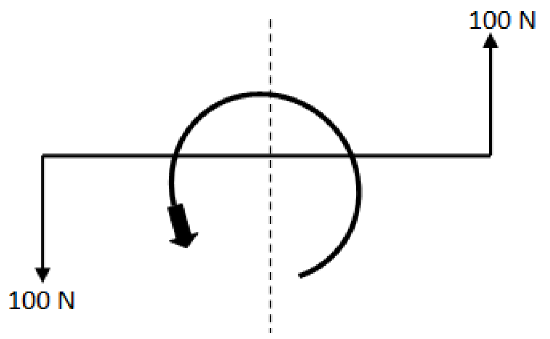
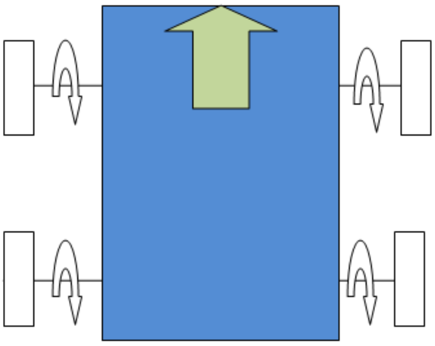
Conventional Vehicle	Vehicle with the proposed gearbox
	
Net torque produced = (100-50)d = 50d N-m	Net torque produced = (100+100)d = 10000d N-m

Table 2

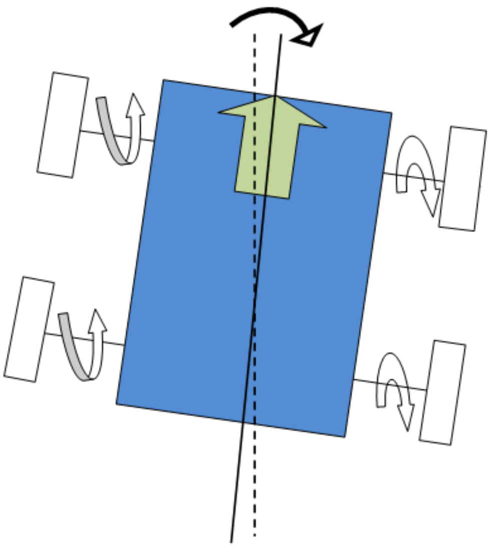
3.3 Comparison between torque produced with and without the proposed gearbox

It can be seen from the above assumed comparison that the torque produced in case of the vehicle with the proposed gearbox will dwarf the torque produced in case of the conventional vehicle . This gargantuan torque produced will turn the vehicle with near zero radius .

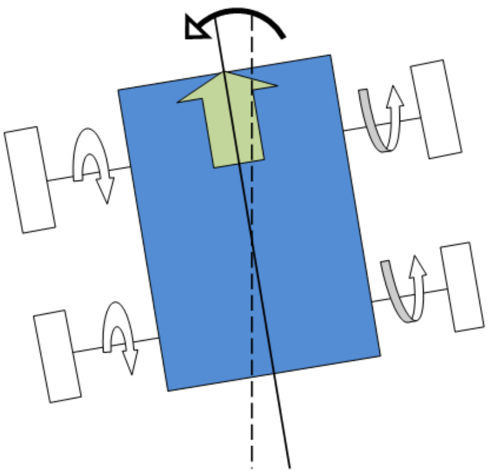
The turning of the vehicle with the proposed gearbox is shown in the following diagrams . The rotation of the axles corresponding to the turns are also shown . When the vehicle is moving forward all the axles rotate in the same direction . When the vehicle turns right , the right set of wheels rotate in the clockwise direction (as seen from the left side) and the left set of wheels rotate in the anti-clockwise direction (as seen from the left side) . When the vehicle turns left , the left set of wheels rotate in the clockwise direction (as seen from the left side) and the right set of wheels rotate in the anti-clockwise direction (as seen from the left side) .



(a) When the vehicle is moving forward

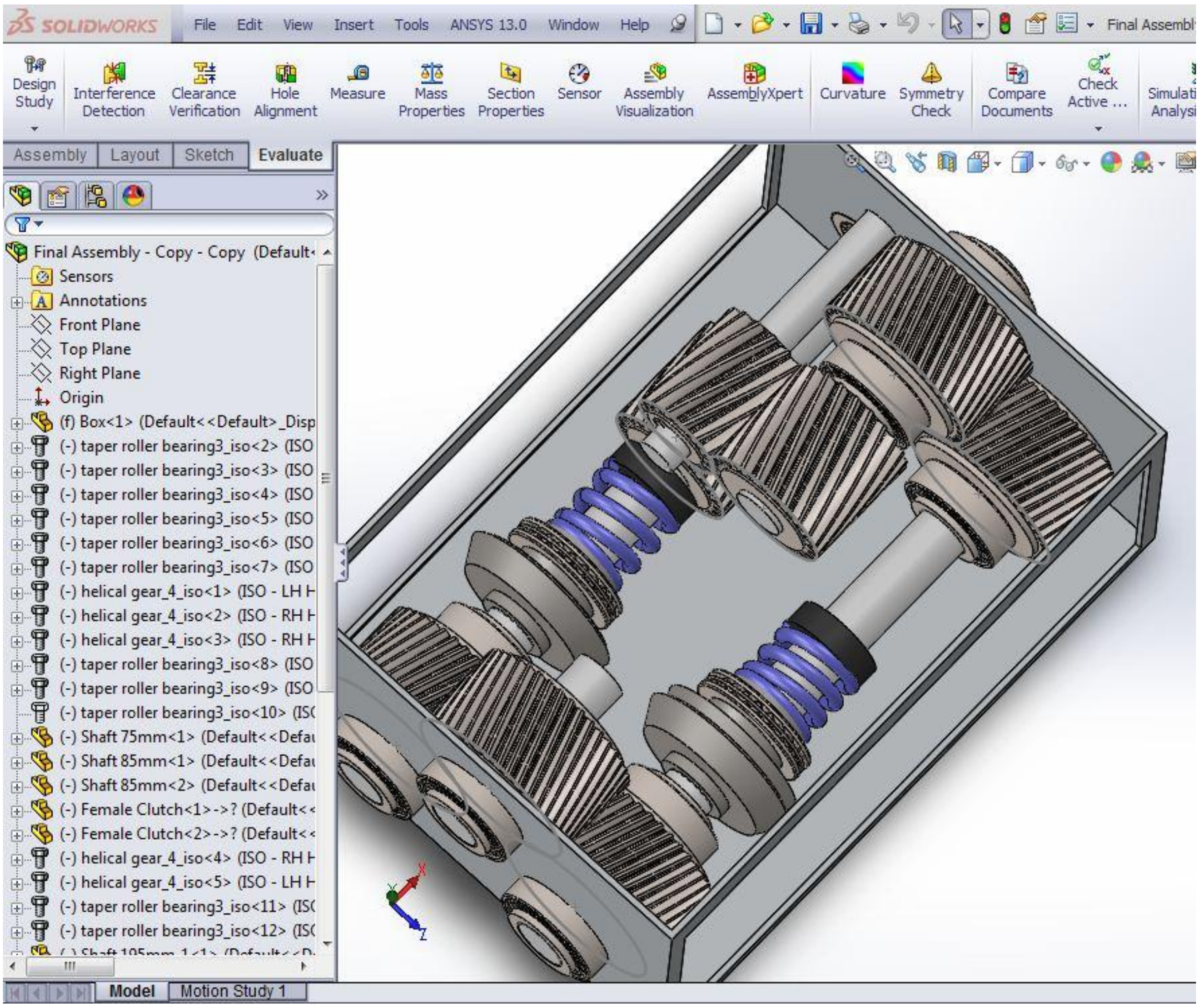


(b) When the vehicle is turning right



(c) When the vehicle is turning left

Fig.6



<u>Screenshot of gear box assembly completely designed in Solidworks.</u>

CHAPTER-4

DESIGN OF COMPONENTS

Introduction

Gearboxes can be engineered to allow gear ratio changes to enable output shaft speed while keeping the input speed and torque at the same value . The primary advantage for using a gearbox for changing speed is to enable the full power to be transmitted at the different speeds . Electric motors and other prime movers are rated for maximum torque at he optimum speed . If the speed is reduced using electronic controls the resulting developed torque is not proportionally increased .

Gearboxes also allow the input shaft and the output shaft to be in different directions .

Gearbox design features

The design of gearbox includes the following features :

- Input and output shaft relative position and orientation
- Support of external forces on shafts
- Design and rigidity of casing
- Type, dimensions and strength of gears
- Method of changing speed/direction if required
- Design and strength of gearshafts
- Gearbox bearings
- Lubrication
- Noise and vibration
- Couplings to shaft
- Maintenance provisions

The components required to be designed for use in the proposed gearbox –
1. Gears (helical gears)
2. Conical Clutch
3. Shaft
4. Keys
5. Relief Spring

BASIC ASSUMPTION

We have taken Mahindra XUV 500 as a reference to design the gearbox .
The specifications of the above said vehicle used for designing the gearbox are –
1. Maximum gross torque : 330 N-m @ 1600-2800 rpm
2. Maximum gross power : 103 kw @ 3750 rpm

CHAPTER-5

DESIGN OF HELICAL GEARS

5.1 Helical Gears

Fig.7

A **gear** is a rotating machine part having cut *teeth*, or *cogs,* which *mesh* with another toothed part in order to transmit torque. *Helical* or "dry fixed" gears offer a refinement over spur gears. The leading edges of the teeth are not parallel to the axis of rotation, but are set at an angle. Since the gear is curved, this angling causes the tooth shape to be a segment of a helix. The former refers to when the shafts are parallel to each other; this is the most common orientation. In the latter, the shafts are non-parallel, and in this configuration the gears are sometimes known as "skew gears". The angled teeth engage more gradually than do spur gear teeth, causing them to run more smoothly and quietly. With parallel helical gears, each pair of teeth first make contact at a single point at one side of the gear wheel; a moving curve of contact then grows gradually across the tooth face to a maximum then recedes until the teeth break contact at a single point on the opposite side. In spur gears, teeth suddenly meet at a line contact across their entire width causing stress and noise. Spur gears make a characteristic whine at high speeds. Whereas spur gears are used for low speed applications and those situations where noise control is not a problem, the use of helical gears is indicated when the application involves high speeds, large power transmission, or where noise abatement is important. The speed is considered to be high when the pitch line velocity exceeds 25 m/s.

Helical gears applied in the industries come in diverse weight and sizes. There are many different types of helical gears which can be used in different places. Therefore, the application of helical gears is diverse, and we cannot give a specific list. We can see that mechanical industry has a quick development. It is no doubt that the market demands for helical gears that are used in industrial applications are increasing. Obviously, the application of helical gears is quite large. For example, they can be used in fertilizer industry, railway industry, printing industry, and earth moving industry, etc. Except for these industries, there are also many others where helical gears can play a rather important role. For instance, helical gears like spur, helical, and planetary form an integral part of the machinery and vehicles used in the earth moving industry. Second, the railway industry railways are indispensable in our society. In railways, there is a wide range of helical gears and equipment used in different railway applications. For the special requirements of the railways, the specialized helical gears are brought into use. They are applied to achieve the certain purposes .

5.2 Advantages of Gear Technology

The advantages of gear technology over other transmission means are:

- Gear technology gives positive drives and constancy of speed ratio without any slippage
- In Gear technology, the drive is very compact due to short centre distances in such drives.
- Gear technology has high efficiency, service , and simple operation.
- Gear technology drives are capable of driving loads subjected to shock at speeds up to 20 m/s
- Maintenance of gear technology drives is inexpensive and if properly lubricated and operated, gear drives have the longest service life compared to other drives.
- Gear technology can be used where precise timing is desired.
- Gear technology can drive much heavier loads than other drives.
- Gear drives can be used for a wide range of transmitted power .

CALCULATION:-

5.3 GEAR 1 (d=60mm)

Material – AISI A11

Ultimate yield stress , $\boldsymbol{\sigma_{ut}}$	= 5205 MPa
Module , **m**	= 1.5 mm
Maximum torque, $\mathbf{T_{max}}$	= 330 N-m @2200 rpm
Rated torque, $\mathbf{T_{rated}}$	=262.28 N-m @3750 rpm

Force:-

Tangential Force

Maximum tangential force, $\mathbf{P_t}$=11000N

Rated tangential force, $\mathbf{P`_t}$ =8742.67N

Calculation of dynamic loading

Service Factor, $\mathbf{C_s}$ $= \frac{P_t}{P'_t}$ = 1.25

Pitch Line Velocity,v $= \frac{\pi DN}{60000}$ = 6.911 m/s

Velocity Factor, C_v $= \frac{5.6}{(5.6+v^{0.5})}$ = 0.681

Effective tangential force, $P_{eff} = \frac{C_s \times P_t}{C_v}$ = 20191 N

Lewis Equation:-

$(FOS \times P_t) \leq (m \times b \times \sigma_b \times Y)$

$(1.5 \times 20191) \leq (1.5 \times 35 \times 1735 \times 0.389)$

$30286.5 < 35433$

Therefore the design is safe.

Virtual Spur Gear (α = 20 degree)

Module, **m** = 1.5mm

Addendum Diameter , $\mathbf{d_A}$ = 60+3 = 63 mm

Virtual Addendum diameter, $\mathbf{d_A}$ $= \frac{63}{cos^2\psi} = 71.35$ mm

Virtual Addendum radius, $\mathbf{r'_a}$ = 35.675 mm

Virtual Pitch Diameter, $\mathbf{d'}$ $= \frac{d}{cos^2\psi} = 67.9$ mm

Virtual No of Teeth, **z´** $= \frac{D}{m_n \, cos^2\psi}$ = 45.29

Virtual Circular Pitch, $\mathbf{p_C}$´ $= \frac{\pi D`}{Z`}$ = 4.71 mm

Length Of path of contact,

$$\boldsymbol{Lp} = \sqrt{{R'_a}^2 - R'^2\cos(\alpha_n)^2} + \sqrt{{r'_a}^2 - r'^2\cos(\alpha_n)^2} - (R' + r')\sin(\alpha_n) = 39.82 - 21.54$$

$= 18.28$mm

Length of arc of contact, $\boldsymbol{L_a} = \frac{L_p}{\cos\alpha_n} = 19.45$

Contact Ratio, **k** $= \frac{L_A`}{P_c} = 4.129$

5.4 GEAR 2 (d=44mm)

Material − AISI A11

Ultimate yield stress , $\boldsymbol{\sigma_{ut}}$ = 5205 MPa

Module , **m** = 1.5 mm

Maximum torque, $\mathbf{T_{max}}$ = 330 N-m @ 2200 rpm

Rated torque, $\mathbf{T_{rated}}$ = 262.28 N-m @ 3750 rpm

Force:-

Tangential force max, $\mathbf{P_t}$ = 15000 N

Tangential force rated, $\mathbf{P'_t}$ = 11921.82 N

<u>Calculation of dynamic loading</u>

Service Factor:-

$C_s = \frac{P_t}{P'_t}$ =1.25

Pitch Line Velocity, **v** $= \frac{\pi DN}{60000} = 5.068$ m/s

Velocity Factor, $\mathbf{C_v}$ $= \frac{5.6}{(5.6+v^{0.5})} = 0.71$

Effective tangential force, $\mathbf{P_{eff}} = \frac{C_s \times P_t}{C_v} = 26408.45$ N

Lewis Equation:-

$(FOS \times P_t) \leq (m \times b \times \sigma_b \times Y)$

$1.5 \times 26408.45 \leq 2 \times 46 \times 1735 \times 0.289$

$39612.675 \leq 52674.6$

Therefore the design is safe.

Virtual Spur Gear (α = 20 degree)

Module, **m** = 2 mm

Addendum Diameter, $\mathbf{d_A}$ = 44+4 = 48mm

Virtual Addendum diameter, $\mathbf{d'_a}$ $= \frac{48}{cos^2\psi} = 54.36$ mm

Virtual Addendum radius, $\mathbf{r'_a}$ = 27.18 mm

Virtual Pitch Diameter, **d´** $= \frac{d}{\cos^2\psi} = 49.83$mm

Virtual No of Teeth, **z´** $= \frac{D}{m_n \, cos^2\psi} = 24.91$

Virtual Circular Pitch, $\mathbf{p_C'}$ $= \frac{\pi D`}{Z`} = 6.3$

5.5 GEAR 3 (d = 30 mm)

Material − AISI A11

Ultimate yield stress , $\boldsymbol{\sigma_{ut}}$ = 5205 MPa

Module , **m** = 1.5 mm

Maximum torque, $\mathbf{T_{max}}$ = 330 N-m @2200 rpm

Rated torque, $\mathbf{T_{rated}}$ =262.28 N-m @3750 rpm

Force:-

Tangential force max, $\mathbf{P_t}$ = 22000 N

Tangential force rated, $\mathbf{P'_t}$ = 17485.33 N

Calculation of dynamic loading

Service Factor, $\mathbf{C_s}$ $= \frac{P_t}{P'_t}$ = 1.25

Pitch Line Velocity, **v** $= \frac{\pi DN}{60000}$ = 3.46 m/sec

Velocity Factor, $\mathbf{C_v}$ $= \frac{5.6}{(5.6+v^{0.5})}$ = 0.75

Effective tangential force, $\mathbf{P_{eff}}$ $= \frac{C_s \times P_t}{C_v}$ = 36666.67 N

Lewis Equation:-

$(FOS \times P_t) \leq (m \times b \times \sigma_b \times Y)$

$1.25 \times 36666.67 \leq 2 \times 46 \times 1735 \times 0.289$

$45833.5 < 46130.18$

Therefore the design is safe.

Virtual Spur Gear (α=20 degree)

Module, **m** = 2 mm

Addendum Diameter, $\mathbf{d_A}$ = 30+4 = 34mm

Virtual Addendum diameter, $\mathbf{d'_a}$ $= \frac{34}{cos^2\psi}$ = 38.5mm

Virtual Addendum radius, $\mathbf{r'_a}$ = 19.25

Virtual Pitch Diameter, d´ $= \frac{d}{\cos^2\psi}$ = 33.97 mm

Virtual No of Teeth, **z´** $= \frac{D}{m_n \, cos^2\psi}$ = 16.987

Virtual Circular Pitch , $\mathbf{P_c}$ $= \frac{\pi D`}{Z`}$ = 6.3

For gear 2 (d = 44 mm) & gear 3 (d = 30 mm) in mesh

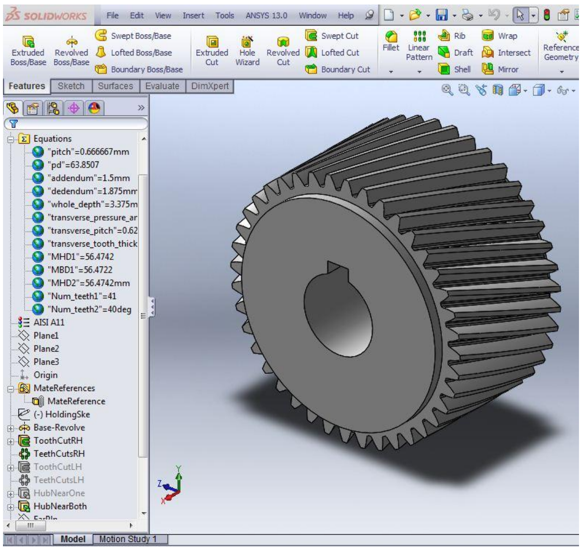
Length Of path of contact

$$\boldsymbol{Lp} = \sqrt{{R'_a}^2 - R'^2\cos(\alpha_n)^2} + \sqrt{{r'_a}^2 - r'^2\cos(\alpha_n)^2} - (R' + r')\sin(\alpha_n) = 13.806 +$$

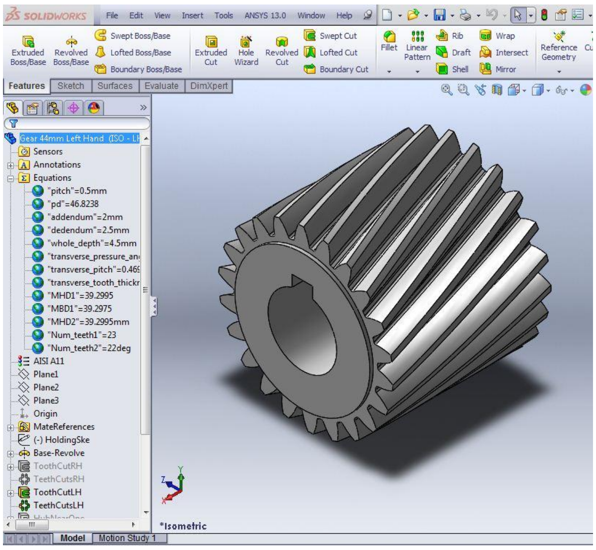
10.762 – 14.33 = 10.28mm

Length of arc of contact, $\mathbf{L_a}$ $= \frac{L_p}{\cos\alpha_n}$ = 10.894

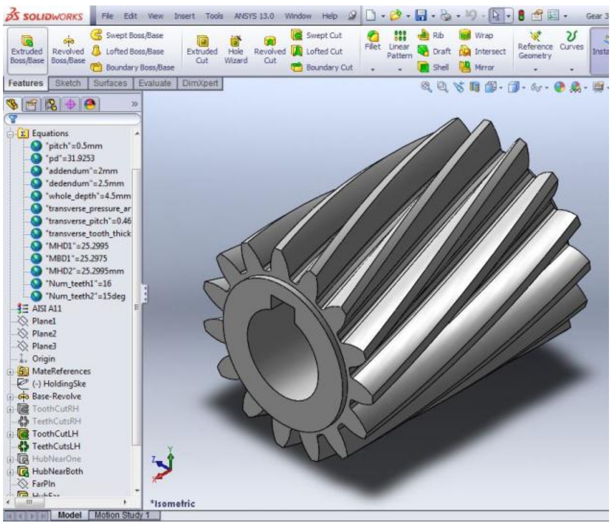
Contact Ratio, **k** $= \frac{L_A}{P_c}$ = 1.72



Helical gear 60mm pitch dia.



Helical gear 44mm pitch dia.



Helical gear 30mm pitch dia.

CHAPTER-6

DESIGN OF CONICAL CLUTCH

6.1 Conical Clutch

A **cone clutch** serves the same purpose as a disk or plate clutch. However, instead of mating two spinning disks, the cone clutch uses two conical surfaces to transmit torque by friction.

The cone clutch transfers a higher torque than plate or disk clutches of the same size due to the wedging action and increased surface area. Cone clutches are generally now only used in low peripheral speed applications although they were once common in automobiles and other combustion engine transmissions.

They are usually now confined to very specialist transmissions in racing, rallying, or in extreme off-road vehicles, although they are common in power boats. This is because the clutch does not have to be pushed in all the way and the gears will be changed quicker. Small cone clutches are used in synchronizer mechanisms in manual transmissions.

Introduction

The cone clutch is an axially actuated clutch which is able to transmit a relatively high torque for its size compared to a single disk plate clutch of the same dimensions. This results from the wedging action and increased friction area. Cone clutches are not used widely now and are generally used for low peripheral speed applications.

The cone angle α is always above 8^o and is normally between 12^o and 15^o. If the angle is less than this value than the clutch is liable to jam in engagement.

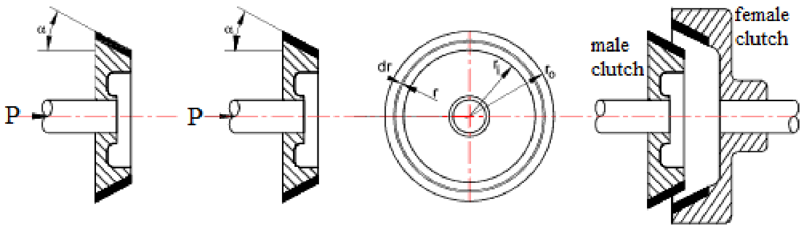
Nomenclature

P	= Axial Applied force kW
M_t	= Clutch Torque (Nm)
μ	= Coefficient of Friction.
D, d	= Rear, Frontal diameter of clutch.
α	=Semi-cone angle

Theory

There are two operating conditions applicable to clutch plates.

- Uniform wear.. Applicable for practical clutch assemblies after period of operation
- Uniform pressure.. Applicable for new clutch plate friction linings.



Cone Clutch

5.2 Calculations

Uniform pressure

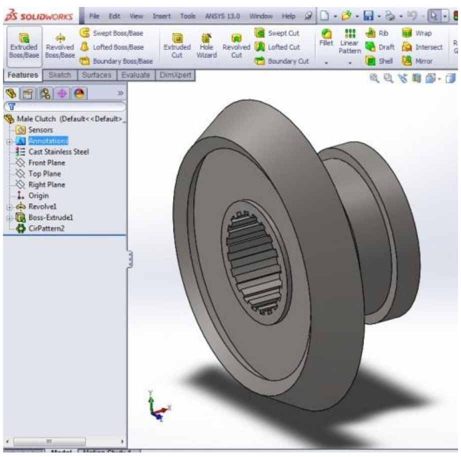
$$Mt = \frac{\mu p}{3\ Sin\ \alpha}\frac{(D^3 - d^3)}{(D^2 - d^2)}$$

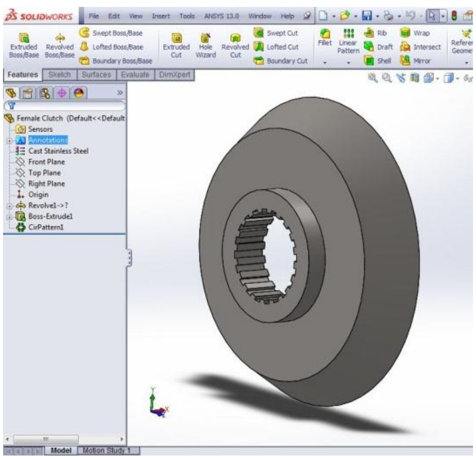
$$330x10^3 = \frac{0.8 \times p}{3\ Sin\ 30}\left(\frac{60^3 - 50^3}{60^2 - 50^2}\right)$$

P = 7479.395 N

Advantages of Conical Clutch

- The normal force acting on the contacting surfaces in case of cone clutches is more than axial force; as compared to in single plate clutch in which the normal force acting on contacting surfaces is equal to the axial force.
- A cone clutch can be rebuilt and reused many times instead of replacing the clutch with a new unit.The materials used in the typical flat clutch are far more lightweight than those used in a cone system. Where the clutch disk in a flat system is made of very thin and lightweight steel, the cone unit is made of heavy, solid steel machined into the correct tolerances.





CHAPTER-7

DESIGN OF SHAFT

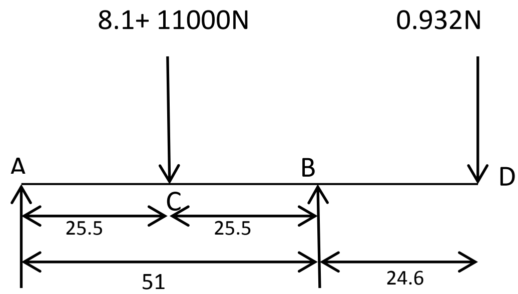
7.1 Left Shaft

Forces on the shaft :-

Tangential Load (P_t) = 11kN

Radial Load(P_r) = $11000 \times \left[\frac{\tan 20}{\cos 20}\right] = 4260.62\ N$ (Helix Angle = 20^0)

Axial Load (P_a) $= 11000\ \times \tan 20 = 4003.67\ N$



Vertical Bending Moment

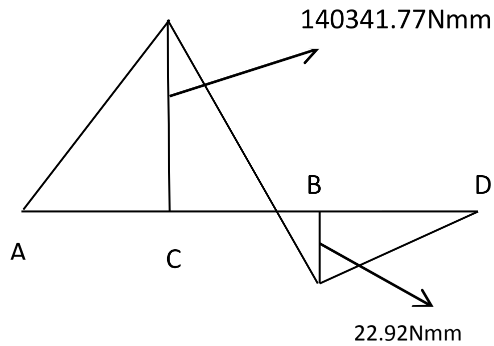
$\sum M_A = 0$

$0.932\ \times 75.6 - Rb \times 51 + 11008.1 \times 25.5 = 0$

$R_B = 5505.43\ N$

$11008.1 + 0.932 = Ra + 5505.43$

$R_A = 5503.6\ N$



Horizontal Bending Moment

$\sum M_A = 0$

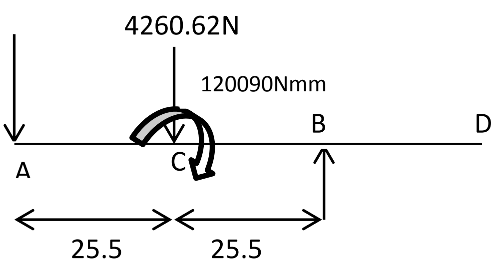
$4260.62\ \times 25.5 + 120090 = Rb\ \times 51$

$R_B = 4451.6N$

$\sum M_B = 0$

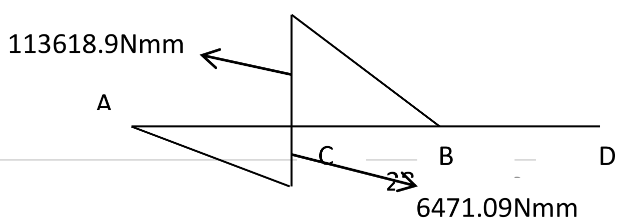
$Ra\ \times 51 + 120090 - 4260.62\ \times 25.1 = 0$

$R_A = -\ 257.813\ N$



Resultant Bending Moment

$M_b = \sqrt{140341.77^2 + 113618.9^2} = 180568.73\ Nmm$



Material properties

S_{ut} = 5205 N/mm^2

S_{yt} = 5171 N/mm^2

$0.3\ Syt = 0.3\ \times 5171 =$ \`1551.3 N/mm^2

$0.18\ Sut = 0.18\ \times 5205 = 936.9$ N/mm^2 (**smaller value**)

$\boldsymbol{\tau}_{\mathbf{max}} = 0.75\ \times 936.9$ (Since there are keyways on the shaft)

= 702.68 N/mm^2

$$d^3 = \frac{16}{\pi\tau} \times \sqrt{(\text{Kb} \times \text{Mb})^2 + (\text{Kt} \times \text{Mt})^2}$$

$$= \frac{16}{\pi \times 702.68}\sqrt{(2.0 \times 180568.73)^2 + (1.5\ \times 120110.1)^2}$$

D = 14.3mm $\approx 20\ mm$

7.2 Right Shaft

Forces on the shaft:-

Tangential Load, $\mathbf{P_t}$ = 15 kN

Radial Load, $\mathbf{P_r}$ $= 15000 \times \left[\frac{\tan 20}{\cos 20}\right] = 5809.935\ N$ (Helix Angle = 20^0)

Axial Load, $\mathbf{P_a}$ $= 15000\ \times \tan 20 = 5459.55\ N$

Vertical Bending Moment

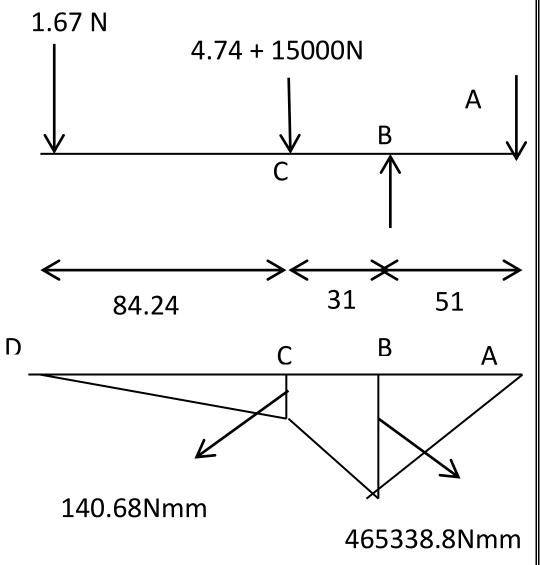
$\sum M_A = 0$

$1.67 \times 166.24 + (4.74 + 15000) \times 82 = \text{Rb} \times 51$

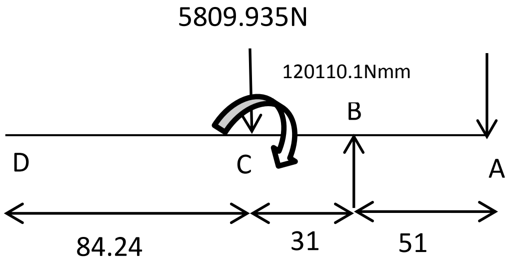
R_B = 24130.7 N

R_A + 24130.7 = 15004.74 + 1.67

R_A = -9124.29 N



Horizontal Bending Moment



$\sum M_A = 0$

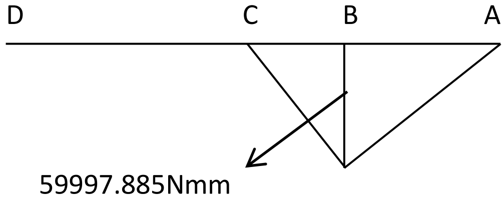
$5809.935 \times 82 - 120110.1 - Rb \times 51 = 0$

$R_B = 6986.36$ N

$\sum M_B = 0$

$5809.935 \times 31 + Ra \times 51 - 120110.1 = 0$

$R_A = -1176.43$ N



Resultant Bending Moment

$M_B = \sqrt{465338.8^2 + 59997.885^2}$

$= 469190.734$ Nmm

Material

$S_{ut} = 5205$ N/mm^2

$S_{yt} = 5171$ N/mm^2

$0.3\,Syt = 0.3 \times 5171 =$ `1551.3 N/mm^2

$0.18\,Sut = 0.18 \times 5205 = 936.9$ N/mm^2 (**smaller value**)

$\tau_{max} = 0.75 \times 936.9$(Since there are keyways on the shaft)

$= 702.68$ N/mm^2

$$d^3 = \frac{16}{\pi\tau} \times \sqrt{(Kb \times Mb)^2 + (Kt \times Mt)^2}$$

$$= \frac{16}{\pi \times 702.68}\sqrt{(2.0 \times 469190.734)^2 + (1.5 \times 120110.1)^2}$$

D $= 19.06\text{mm} \approx 20\ mm$

Hence the diameter of the shaft is taken 20 mm after standardising it .

CHAPTER-8

DESIGN OF KEYS

8.1 Rectangular Sunk Key

Parallel keys or Sunk Keys are the most widely used keys. They have a square or rectangular cross-section. Square keys are used for smaller shafts and rectangular faced keys are used for shaft diameters over 6.5 in (170 mm) or when the wall thickness of the mating hub is an issue. Set screws often accompany parallel keys to lock the mating parts into place. The keyway is a longitudinal slot in both the shaft and mating part.

The key we are designing will transmit torque from shaft to the gears .

D = 20mm

Dimension (10×10) mm (assumed)

Material:- ${}^{65}_{8}C$

Γyp = 740 MPa

FOS = 2

$\sigma_T = \frac{740}{2}$

=> $370 = \frac{2\times T}{d(\frac{h}{2}\times l)} = \frac{2\times 330\times 1000}{20\times 5\times l}$

=> **L = 17.838 mm**

$\tau = \frac{\sigma t}{2} = \frac{740}{4}$

=> $185 = \frac{2T}{d\times b\times l} = \frac{2\times 330\times 1000}{20\times 10\times l}$

=> **L=17.85 mm**

Hence the final dimension of the key is (18×10×10) mm . (l×b×h)

CHAPTER-9

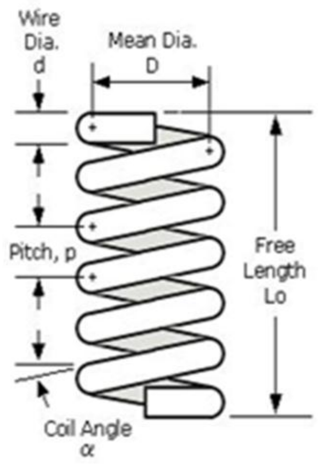
DESIGN OF RELIEF SPRING

The relief spring is integrated in the conical clutch . This spring helps in the engagement of the male and female part of the conical clutch which in turn helps in the engagement /disengagement of the auxiliary shafts from the main shaft .

Axial load on the spring, **P** = 7500 N

Outer diameter of the spring, **D** = 30 mm

Spring stiffness, **k** = 75 N/mm



Material : oil-hardened and tempered steel wire (assumed)

Ultimate tensile strength, $\mathbf{s_{ut}}$ = 1250 N/mm^2

Modulus of rigidity ,**G** = 81370 N/mm^2

Permissible shear stress, τ = 0.3 × $\mathbf{s_{ut}}$

= 375 Mpa

Spring index, **C** = $\frac{D}{d}$ ------------------ (1)

Also , $\tau = k \times \frac{8PC}{\pi d^2}$ ------------------- (2)

From equation (1) & (2)

KC^3 = 200 ------------------- (3)

The Wahl factor is given by :

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

Equation (3) is to be solved by the trial and error method . The values are tabulated in the following way :-

C	K	KC^3
2	1.4425	11.54
3	1.1700	31.62
4	1.0960	70.14
5.5	1.2780	212.6
5.75	1.2650	240.5
5.45	1.2814	207.4

Hence , C = 5.45   (From the above table)

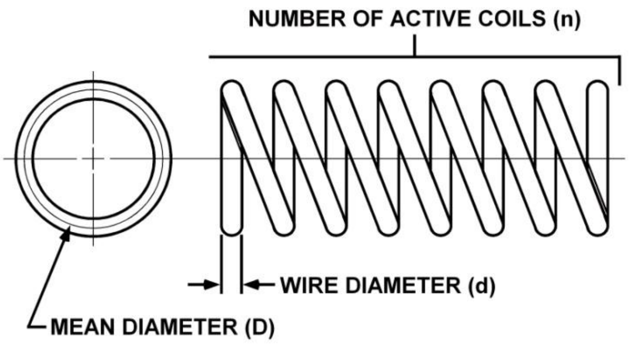
Therefore, wire diameter, **d** $= \dfrac{30}{5.45}$

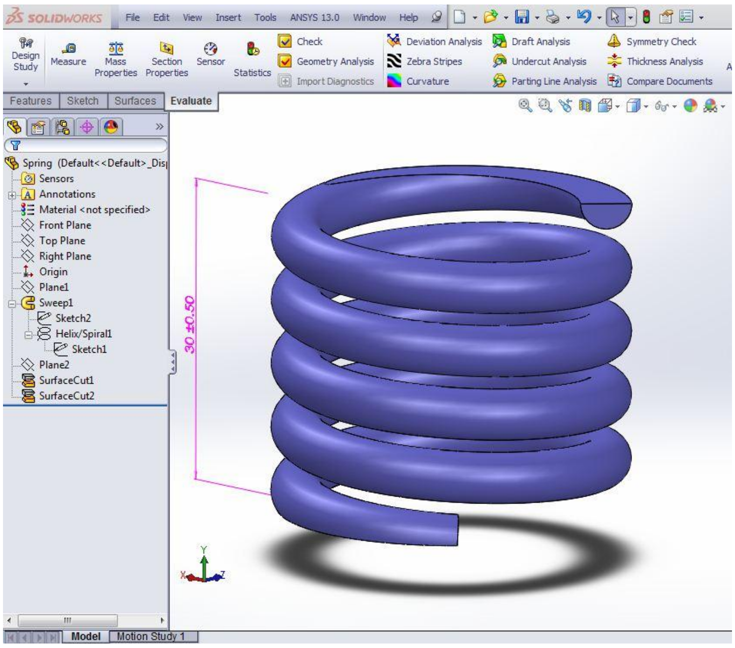
$= 5.5$ mm

Also, $K = \dfrac{Gd^4}{8D^3N}$

$$N = \frac{81370 \times 5.5^4}{8 \times 30^3 \times 75}$$

$= 4.59$ mm $\approx$ 5 **coils**   (where N = Number of coils)





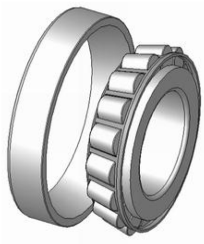
CHAPTER-10

BEARING

10.1 Tapered Bearing

Tapered roller bearings are bearings that can take large axial forces (i.e., they are good thrust bearings) as well as being able to sustain large radial forces.

Description



The inner and outer ring raceways are segments of cones and the rollers are also made with a taper so that the conical surfaces of the raceways and the roller axes if projected, would all meet at a common point on the main axis of the bearing.

A single tapered bearing

This conical geometry is used as it gives a larger contact patch, which permits greater loads to be carried than with spherical (ball) bearings, while the geometry means that the tangential speeds of the surfaces of each of the rollers are the same as their raceways along the whole length of the contact patch and no differential scrubbing occurs.

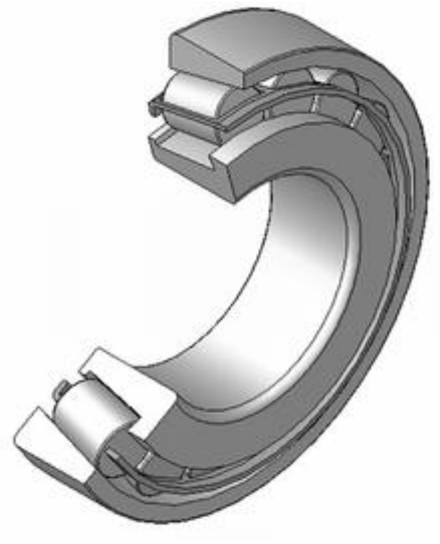
When a roller slides rather than rolls, it can generate wear at the roller-to-race interface, i.e. the differences in surface speeds creates a scrubbing action. Wear will degenerate the close tolerances normally held in the bearing and can lead to other problems. Much closer to pure rolling can be achieved in a tapered roller bearing and this avoids rapid wear.

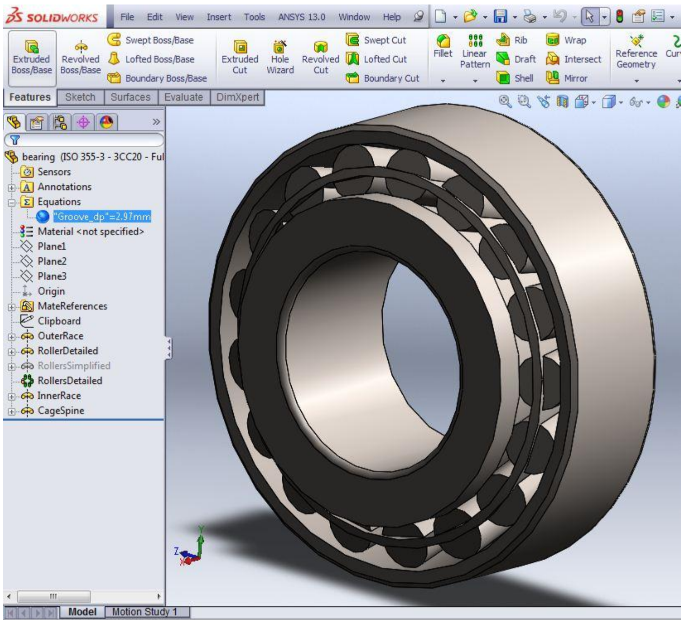
Tapered roller bearings are based on the observation that cones that meet at a point can roll over each other without slipping. In practice, sections of cones(frustum) are used.

The rollers are guided by a flange on the inner ring. This stops the rollers from sliding out at high speed due to their momentum. The larger the half angles of these cones the larger the axial force that the bearing can sustain. Tapered roller bearings are separable and have the following components: *outer ring*, *inner ring*, and *roller assembly* (containing the rollers and a cage). The non-separable inner ring and roller assembly is called the *cone*, and the outer ring is called the *cup*. Internal clearance is established during mounting by the axial position of the cone relative to the cup.

10.2 Applications

In many applications tapered roller bearings are used in back-to-back pairs so that axial forces can be supported equally in either direction. Pairs of tapered roller bearings are used in car and vehicle wheel bearings where they must cope simultaneously with large vertical (radial) and horizontal (axial) forces. Applications for tapered roller bearings are commonly used for moderate speed, heavy duty applications where durability is required. Common real world applications are in agriculture, construction and mining equipment, axle systems, gear box, engine motors and reducers.





Design of tapered roller bearing in Solidworks

CHAPTER-11

CONCLUSION

The above idea is feasible. The idea of having Torque Reversal in civilians vehicle arose due to heavy traffic rise and consequent decrease in road space. The design considerations were made taking into account the condition of Indian roads. The proposed idea can be used very effectively and can be implemented on existing vehicles without major changes to the vehicle itself.

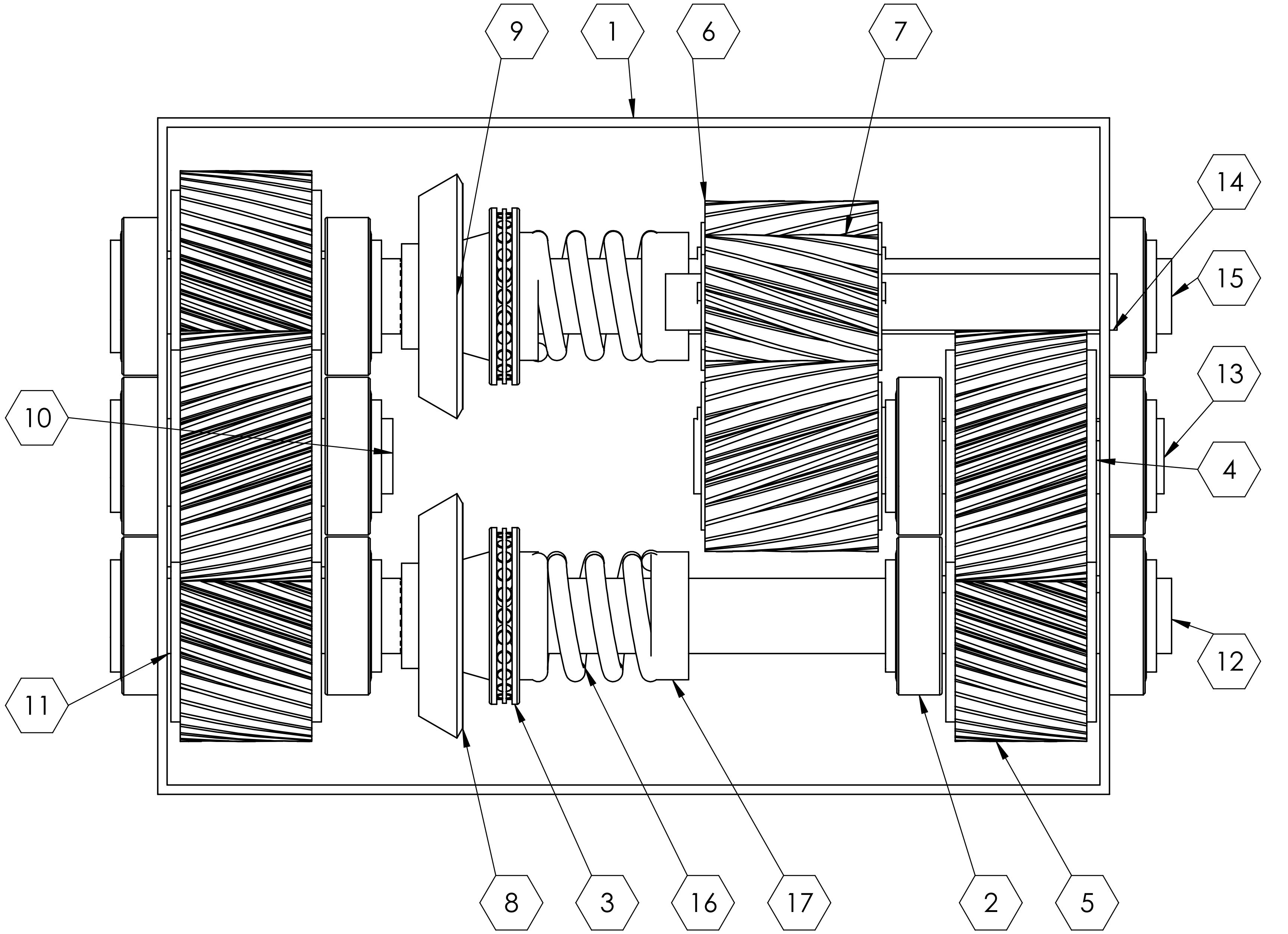
CHAPTER-12

FUTURE SCOPE

We have disregarded many factors which we could have implemented in the designing of the system. These factors have been disregarded to simplify the designing process.

The following lists some of the factors which could have been taken into account while designing:-

- Unbalanced Radial and Axial Force which arises due to the use of helical gears
- Our design was specifically was for 4WD but with some modifications can be used in 2WD
- Cone clutch can be replaced by multi-plate clutch which would require much more detail analysis of the problem
- The shaft has been considered weightless for simplification of design of shaft
- The design is a bit inefficient because we have over-engineered the product. Some weight could be reduced by redesigning the shaft
- We have not taken into account the vibration and noise produced by the gearbox which can be taken into account for future works



ITEM NO.	PART NUMBER	MATERIAL	QTY.
1	Box	Cast Steel	1
2	ISO 355-3 - 3CC20 - Full,DE,AC,Full	(Bearing)Cast Steel	11
3	ISO 104 - 713047 - R,Full,DE,AC,Full	(Bearing) Cast Steel	2
4	ISO - LH Helical gear 1.5M 40T 20HA 20PA 35FW --- 40C70H40L20.0S1	AISI A11	2
5	ISO - RH Helical gear 1.5M 40T 20HA 20PA 35FW --- 40C70H40L20.0S1	AISI A11	3
6	ISO - LH Helical gear 2M 22T 20HA 20PA 46FW --- 22C70H48L20.0S1	AISI A11	2
7	ISO - RH Helical gear 2M 15T 20HA 20PA 46FW --- 15C70H48L15.0S1(30m m)	AISI A11	1
8	Male Clutch	Cast Steel	2
9	Female Clutch	Cast Steel	2
10	Shaft 75mm	AISI A11	1
11	Shaft 85mm	AISI A11	2
12	Shaft 195mm_1	AISI A11	1
13	shaft 125mm	AISI A11	1
14	Shaft 120mm	AISI A11	1
15	Shaft 195mm_2	AISI A11	1
16	Spring Assembly	Steel wire (Tempered)	2
17	Fixed Part		2

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
Bhubaneswar, Orissa, India

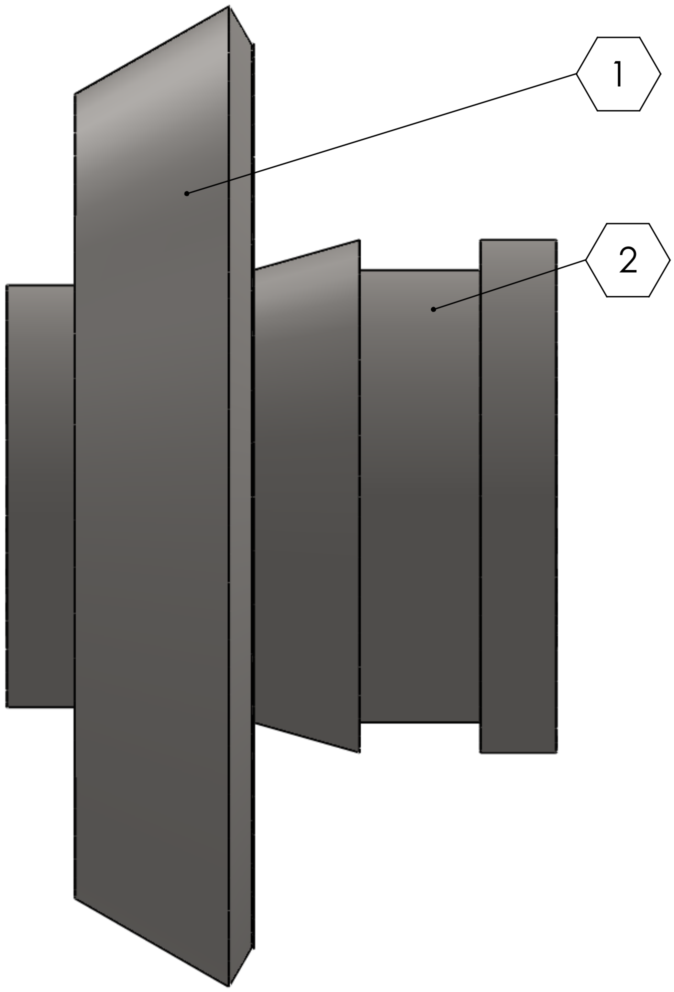
	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL:
				WEIGHT:

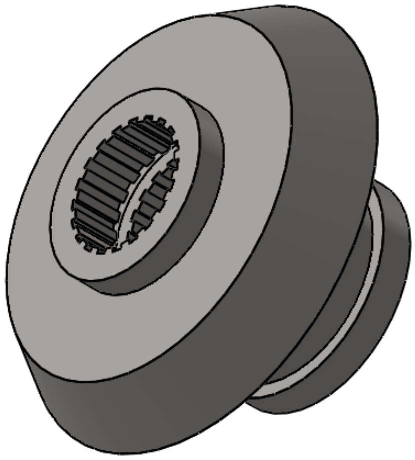
TITLE:

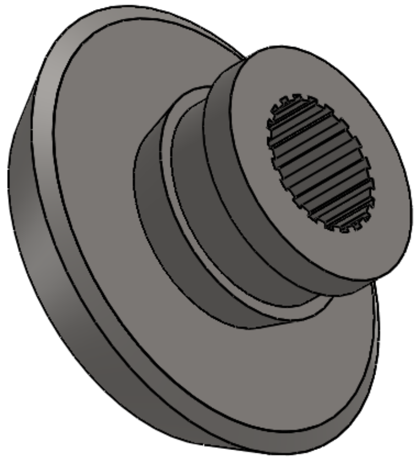
GEAR BOX ASSEMBLY

DWG NO. GB20130305 A3

SCALE:1:1.5 SHEET 1 OF 10







ITEM NO.	PART NUMBER	MATERIAL	QTY.
1	Female Cone	Cast Stailness Steel	1
2	Male Cone	Cast Stailness Steel	1

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL:
				WEIGHT: 277.14 grams

TITLE:

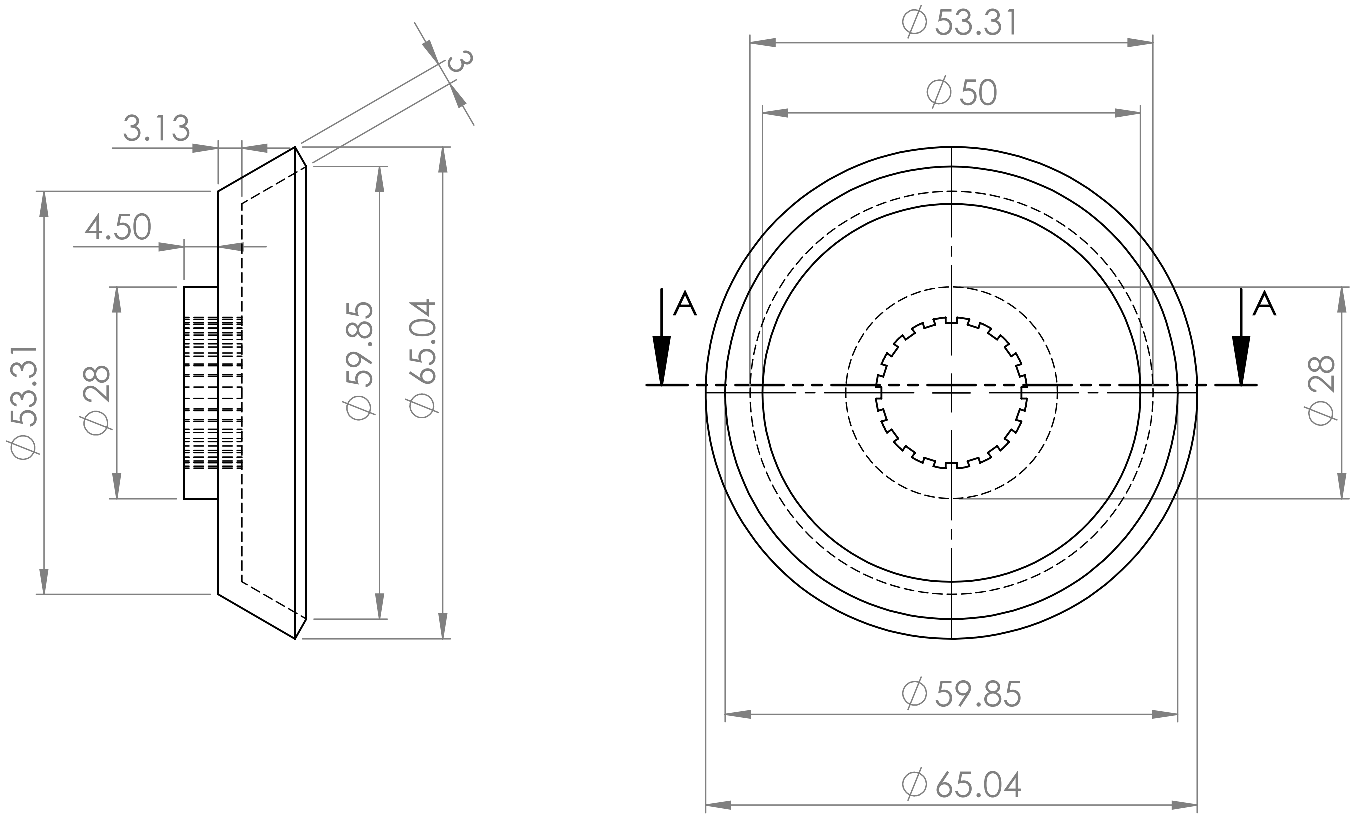
CLUTCH ASSEMBLY

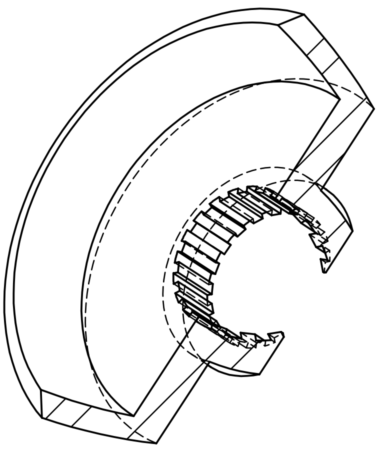
DWG NO. CLTCH01

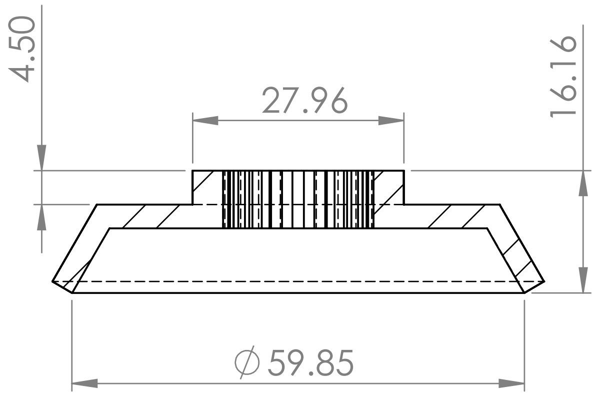
A4

SCALE:1:1

SHEET 2 OF 10 (1/3)







SECTION A-A

SECTION VIEW-VIEW

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL: CAST STAINLESS STEEL
				WEIGHT: 99.45 grams

TITLE:

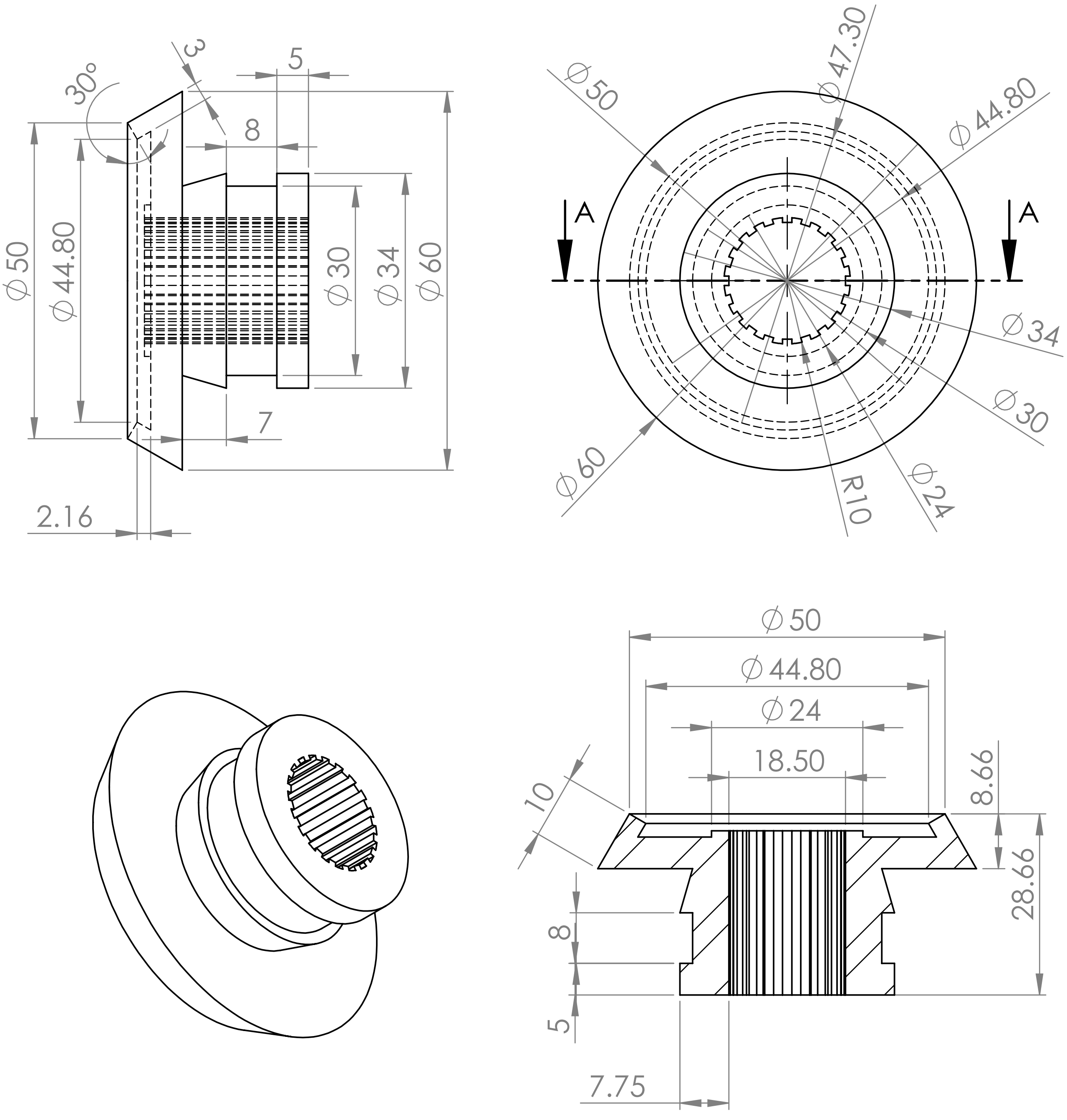
FEMALE CONE

DWG NO. CLTCH201302

A4

SCALE:1:1

SHEET 2 OF 10 (2/3)



SECTION A-A

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE: MALE CONE

MATERIAL: CAST STAINLESS STEEL

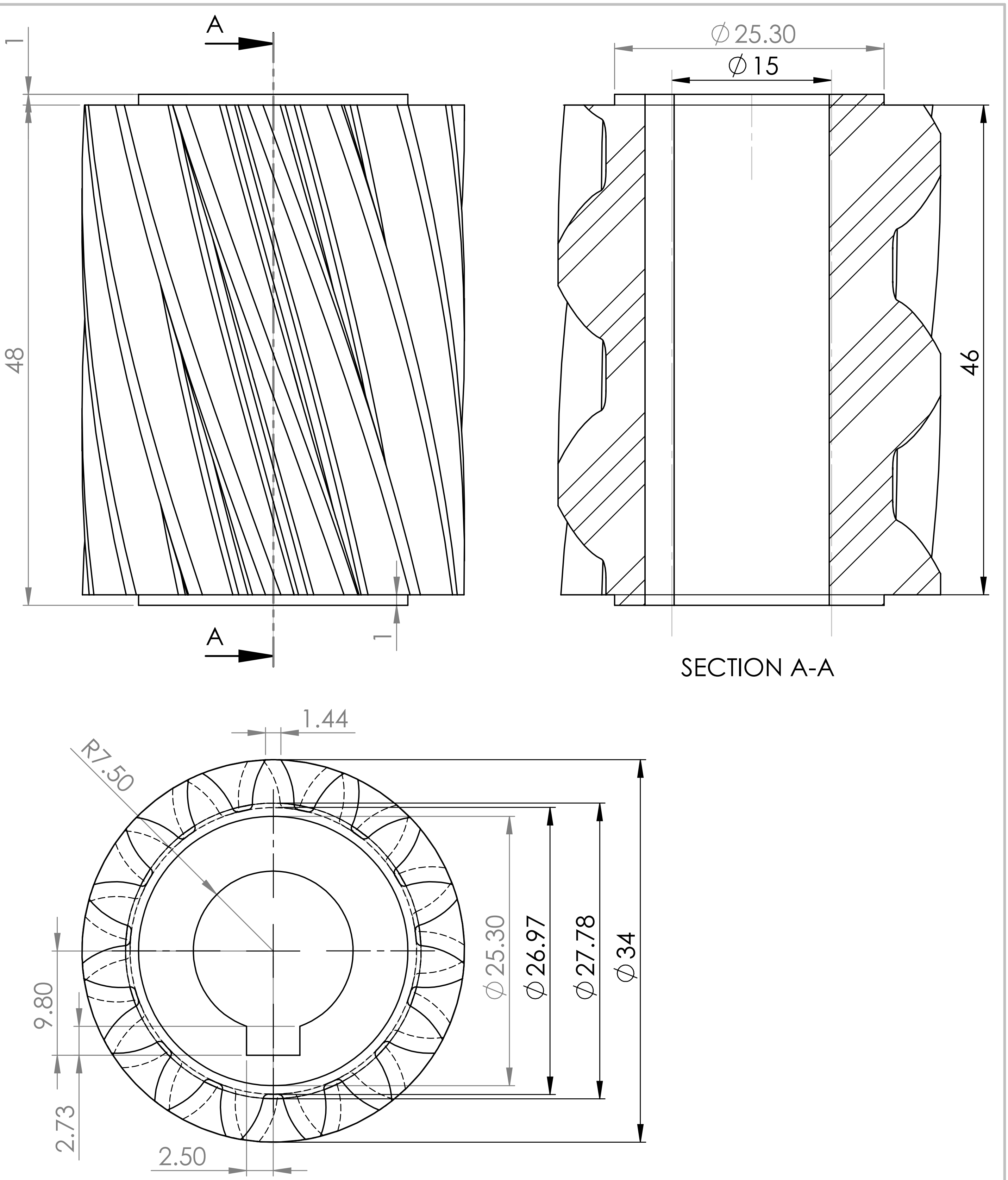
WEIGHT: 177.69 grams

DWG NO. CLTCH201303

A4

SCALE:1:1

SHEET 2 OF 10 (3/3)



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE:
HELICAL GEAR (30 MM)

MATERIAL:
AISI A11

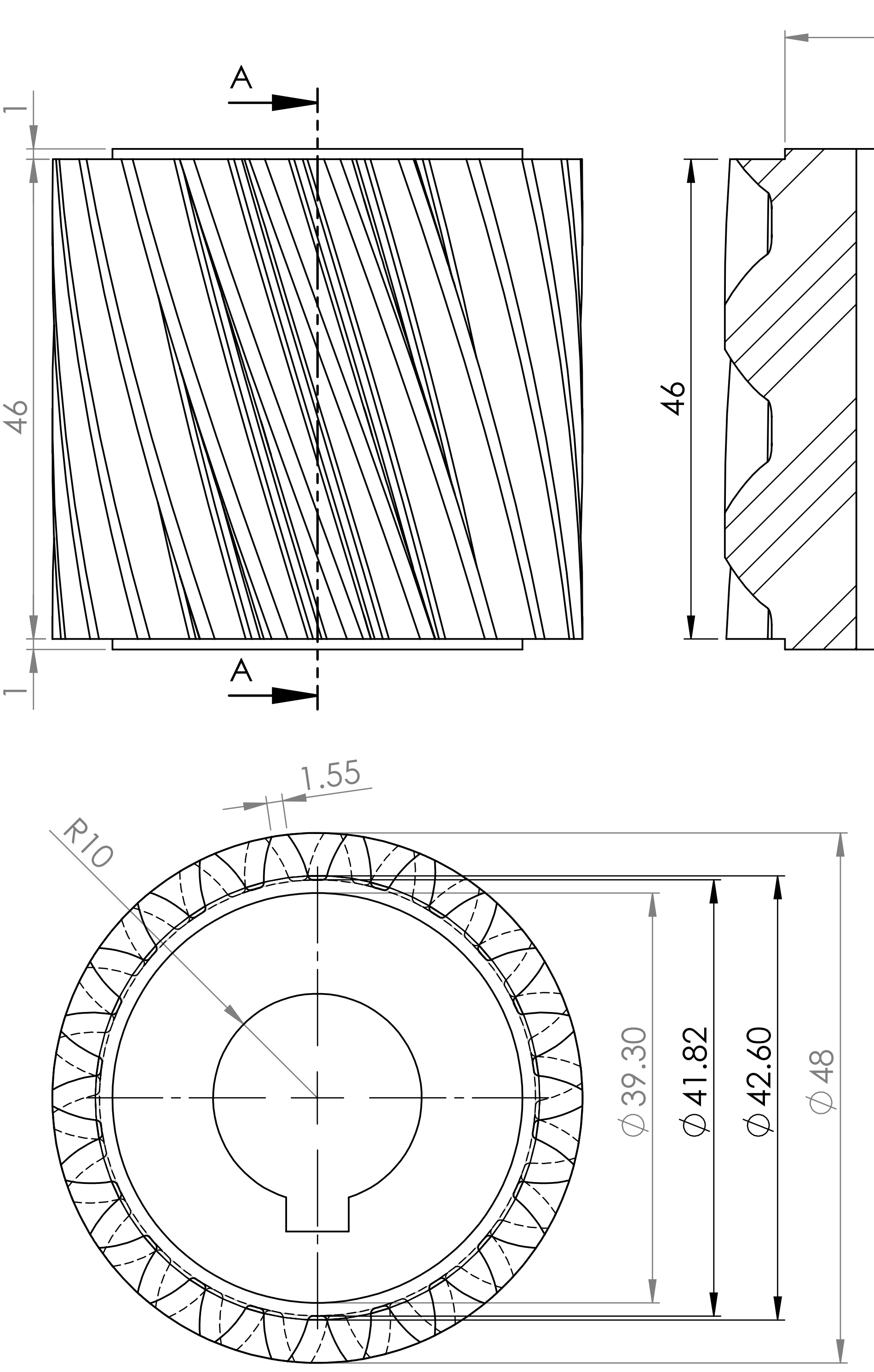
WEIGHT: 210g

DWG NO. 15T46FW30PD

A4

SCALE:2:1

SHEET 3 OF 10



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE:
HELICAL GEAR (44 MM)

MATERIAL:
AISI A11

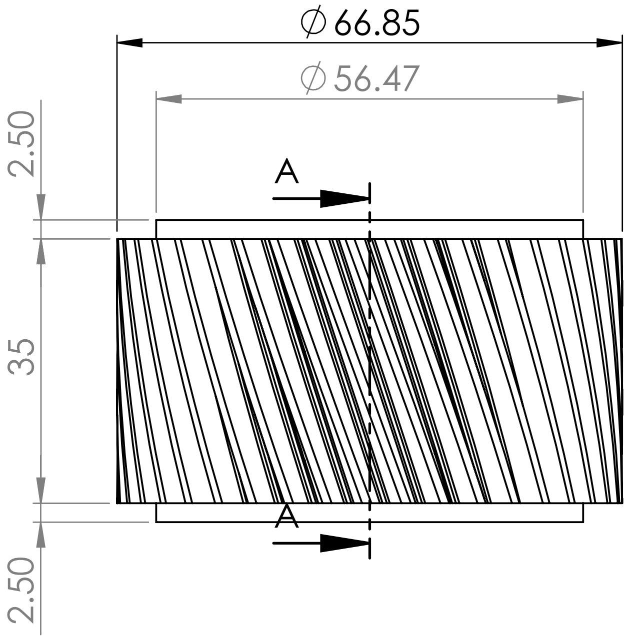
WEIGHT: 483.32g

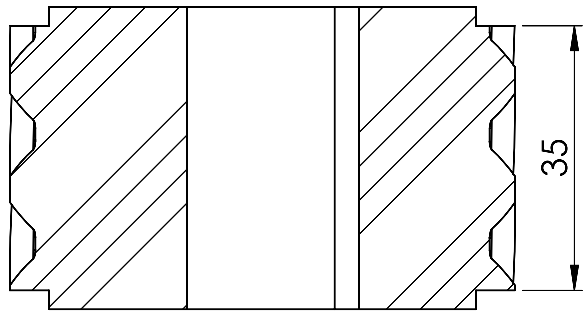
DWG NO.
22T46FW44PD

A4

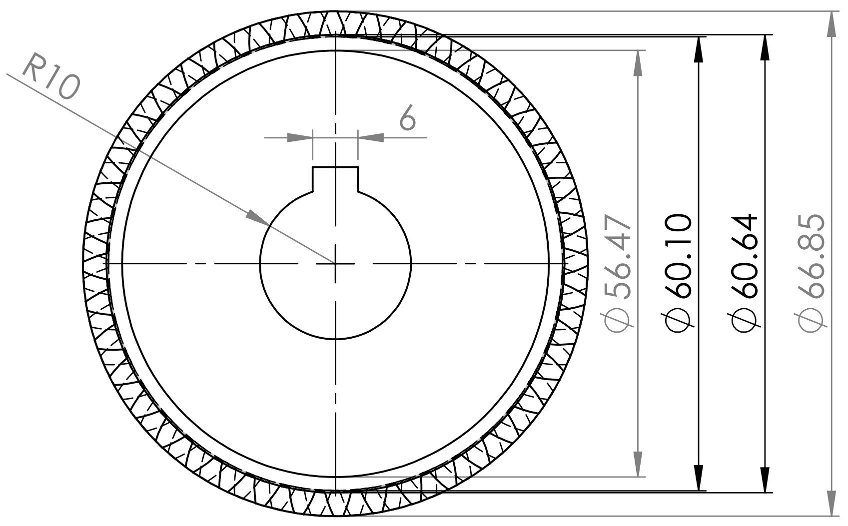
SCALE:1.5:1

SHEET 4 OF 10





SECTION A-A



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

MATERIAL: AISI A11

WEIGHT: 824.73g

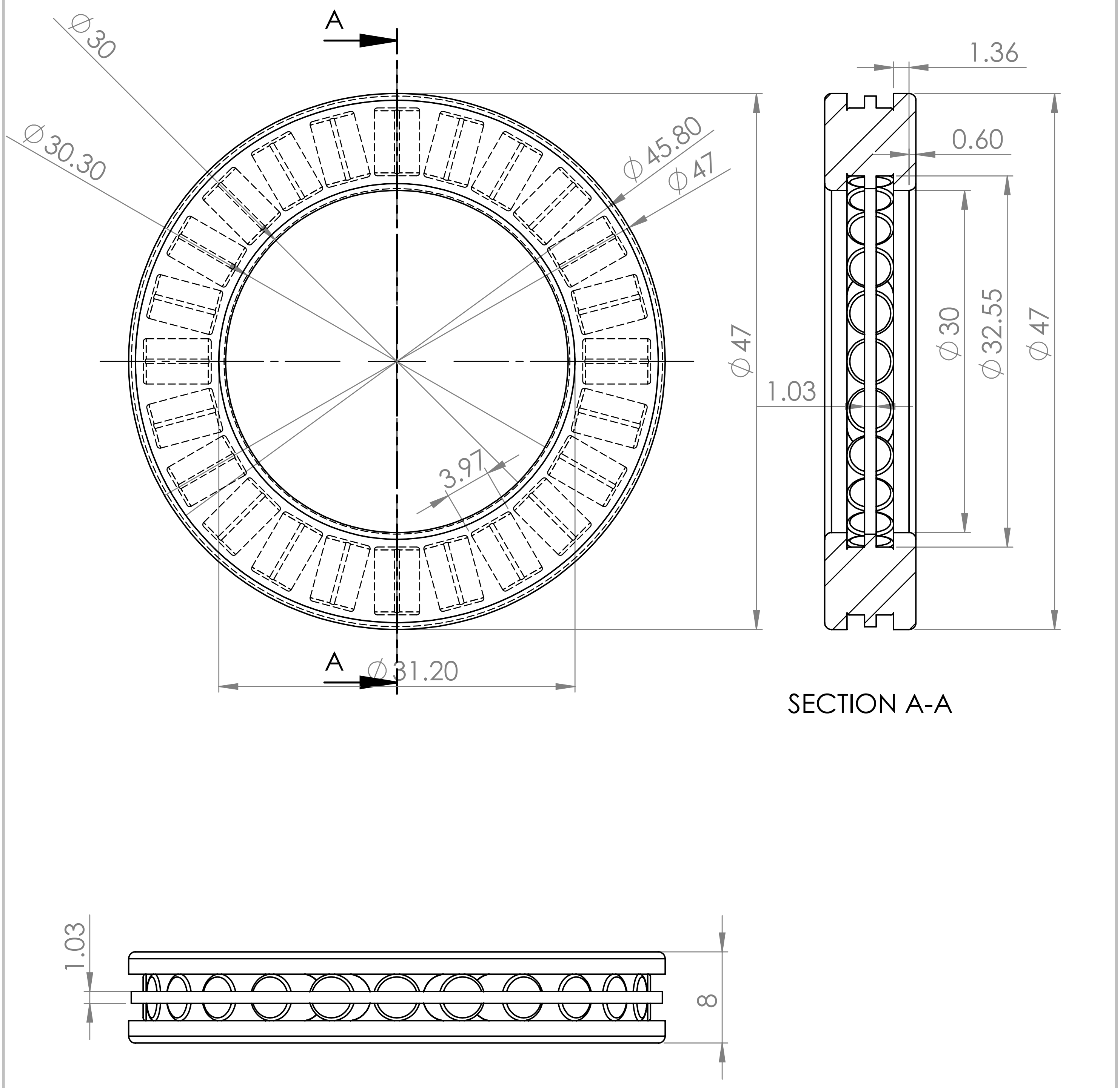
TITLE: HELICAL GEAR (60 MM)

DWG NO. 40T35FW60PD

A4

SCALE:1:1

SHEET 5 OF 10



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE:
ROLLER THRUST BEARING

MATERIAL:
CAST STAINLESS STEEL

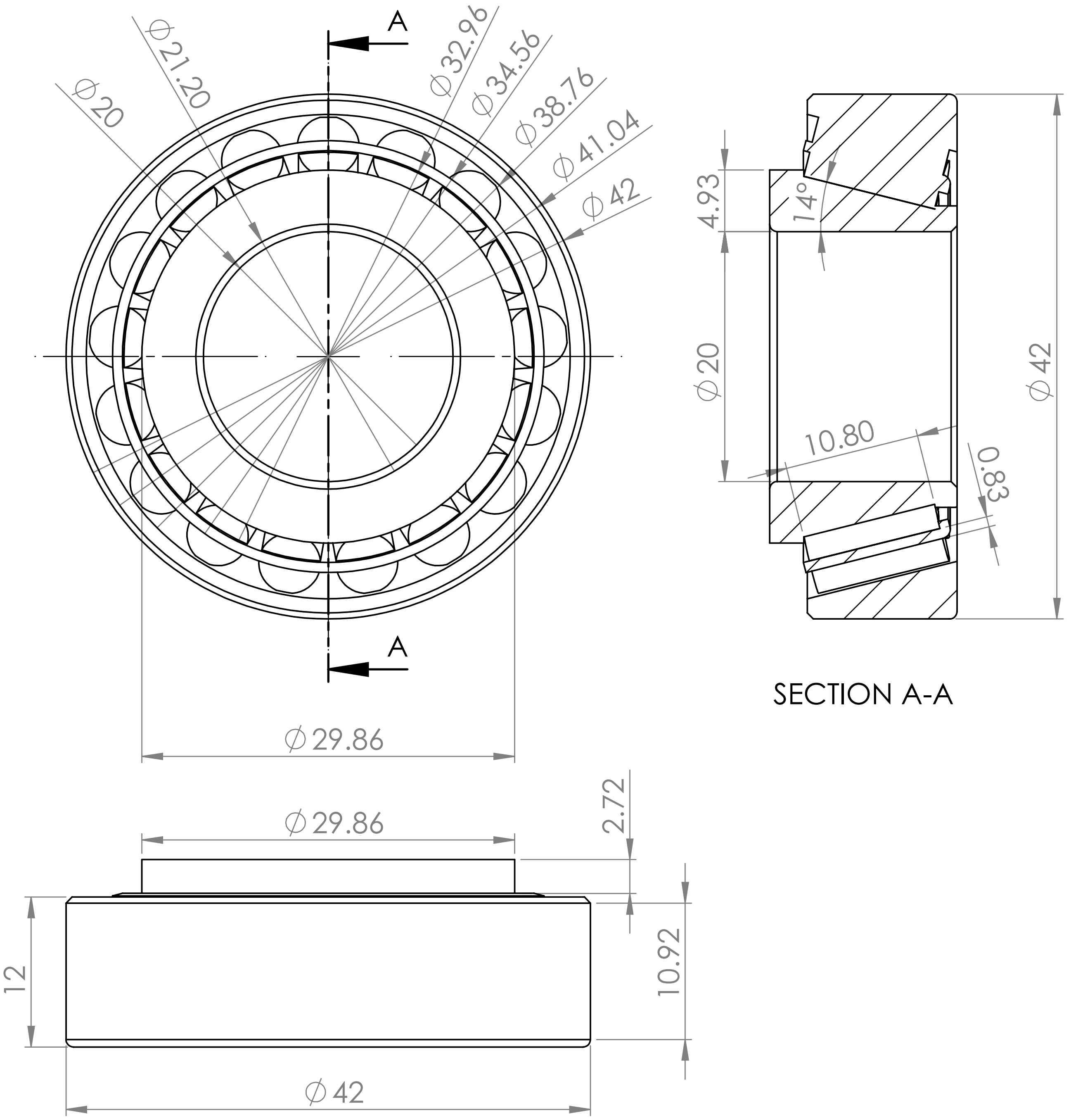
WEIGHT: 48.36 grams

DWG NO. ISO 104 - 713047 - R,Full,DE,AC,Full

A4

SCALE:2:1

SHEET 6 OF 10



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE:

TAPER BEARING

MATERIAL: CAST STAINLESS STEEL

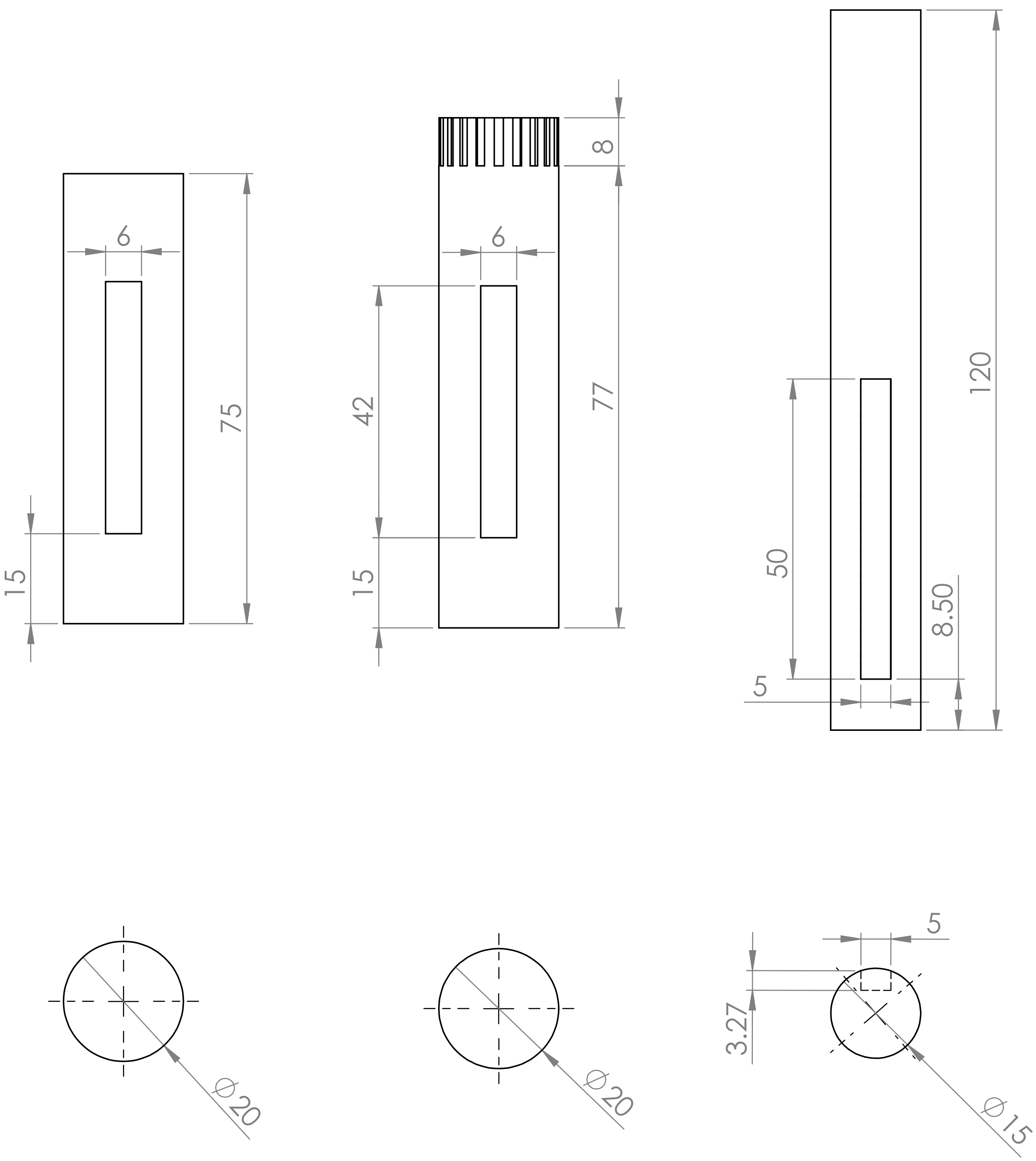
WEIGHT: 93 grams

DWG NO. ISO 355-3 - 3CC20 - Full,DE,AC,Full

A4

SCALE:2:1

SHEET 7 OF 10



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL: AISI A11
				WEIGHT:

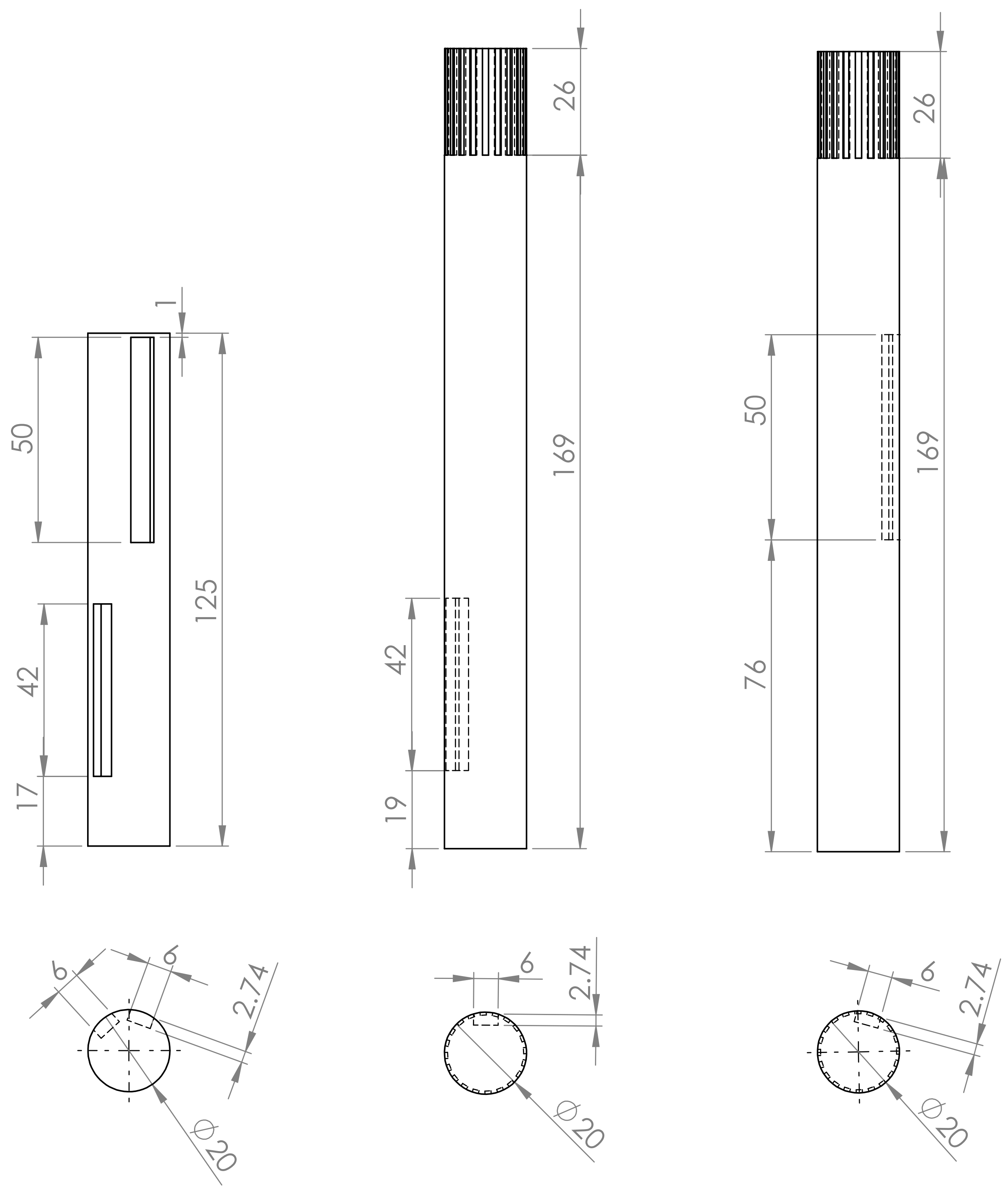
TITLE: SHAFTS-1

DWG NO. SHFT201303

A4

SCALE:1:1

SHEET 8 OF 10



UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				

MATERIAL: AISI A11

WEIGHT:

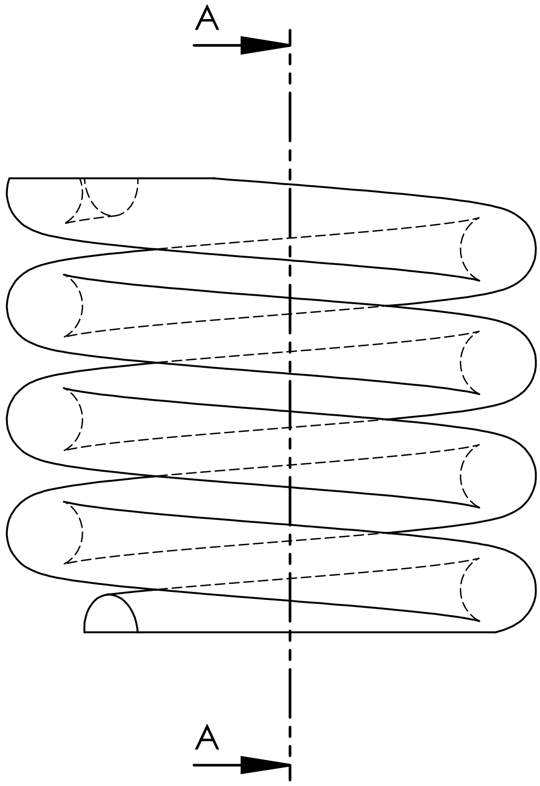
TITLE: SHAFTS-2

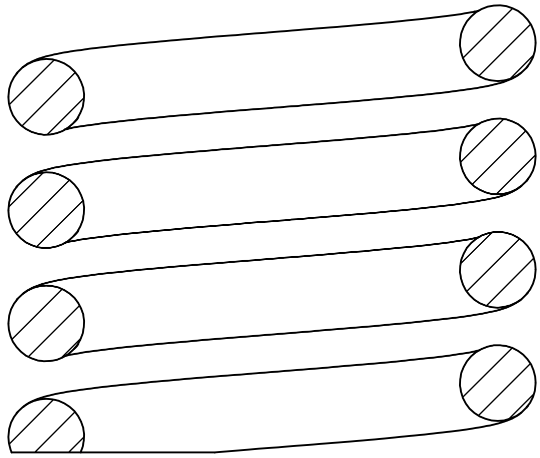
DWG NO. SHFT20130302

A4

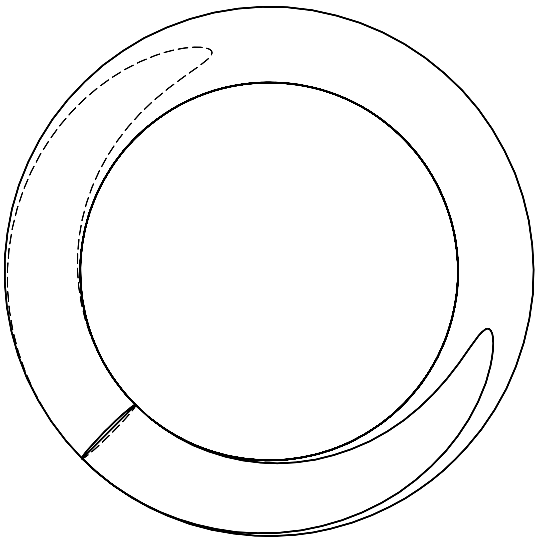
SCALE:1:1.5

SHEET 9 OF 10





SECTION A-A



Specifications:
1. Wire dia= $\varnothing$5 mm
2. Height= 30 mm
3. Pitch Dia.=$\varnothing$30 mm
4. No. of turns=4

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS.
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

KIIT UNIVERSITY
(Declared U/S 3 of UGC Act, 1956)
Bhubaneswar, Orissa, India

	NAME	SIGNATURE	DATE	REVISION
DRAWN	ABHIJEET			
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL: Tempered Steel
				WEIGHT:

TITLE:

SPRING

DWG NO. SPNG201303 | A4

SCALE:2:1 | SHEET 10 OF 10

BIBLIOGRAPHY

- Mechanical Design Data Book- *Shinto Mathew*
- Material Data- *www.matweb.com*
- Design Of Machine Elements – *Dr. V.B. Bhandari*
- Design Data Handbook For Mechanical Engineers- *K. Mahadevan*
- Design Data Handbook- *S. Md. Jalaludeen*
- Theory of Machines- *S. S. Rattan*
- Strength of materials- *S. S. Rattan*
- Mechanism and Machine theory- *Ashok G. Ambekar*
- Bearing catalogue - *http://www.ntnamericas.com/en/website/documents/brochures-and-literature/catalogs/ntn_a-1500-iii_low_res_d1_20091218.pdf*
- Taper roller bearing catalogue - *http://neibearing.com/nbcpdf/new_TechnicalCatalogue.pdf*
- Transmission - *http://en.wikipedia.org/wiki/Transmission_(mechanics)*
- Gearbox - *http://www.scribd.com/doc/31315663/Gear-Box-Design-Report*
- Tank's power transmission - *http://en.wikipedia.org/wiki/Tank*